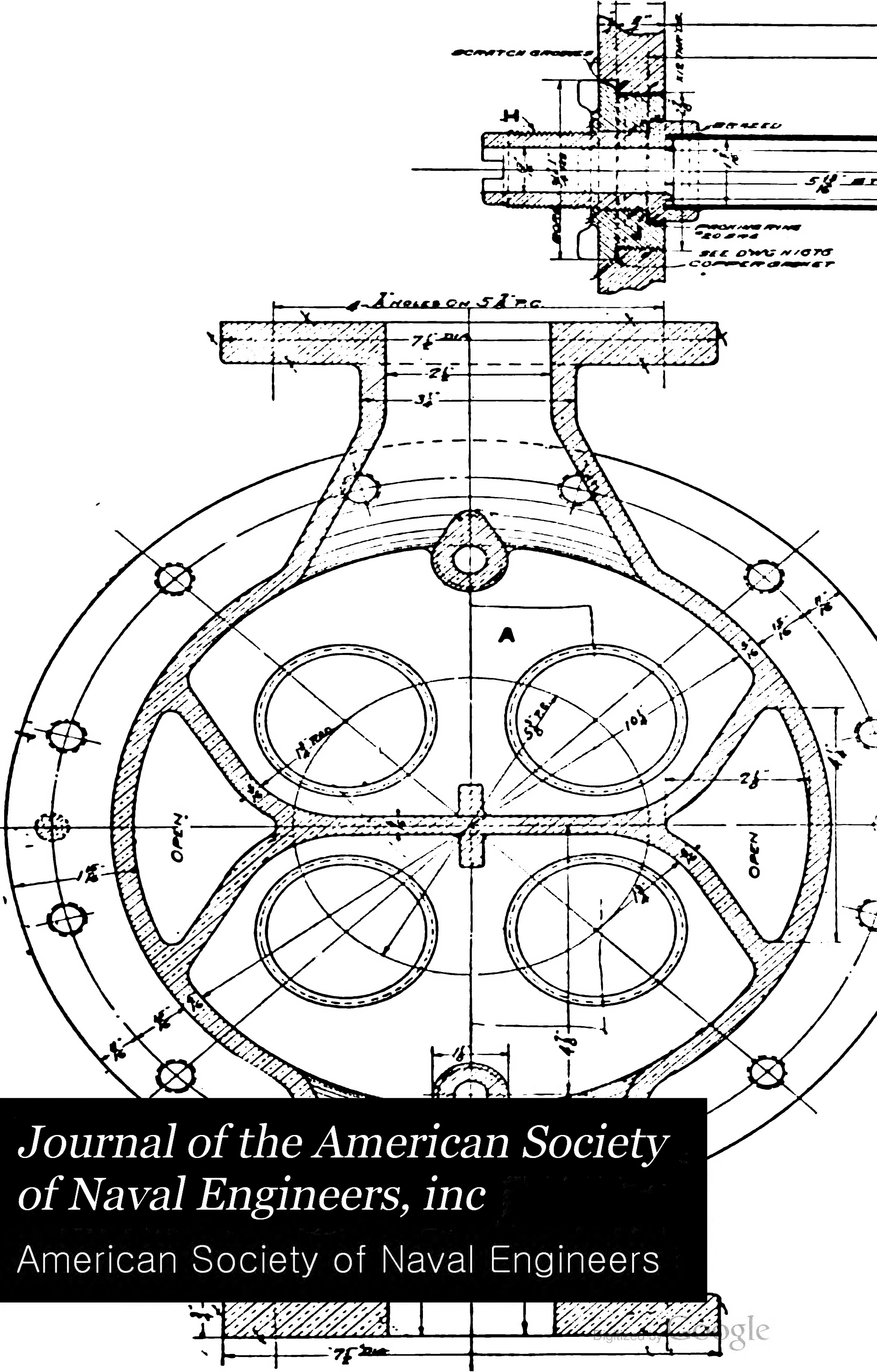


Journal of the American Society of Naval Engineers, inc

American Society of Naval Engineers

JOURNAL

OF THE

American Society of Naval Engineers.

PUBLISHED QUARTERLY

UNDER THE SUPERVISION OF THE COUNCIL.

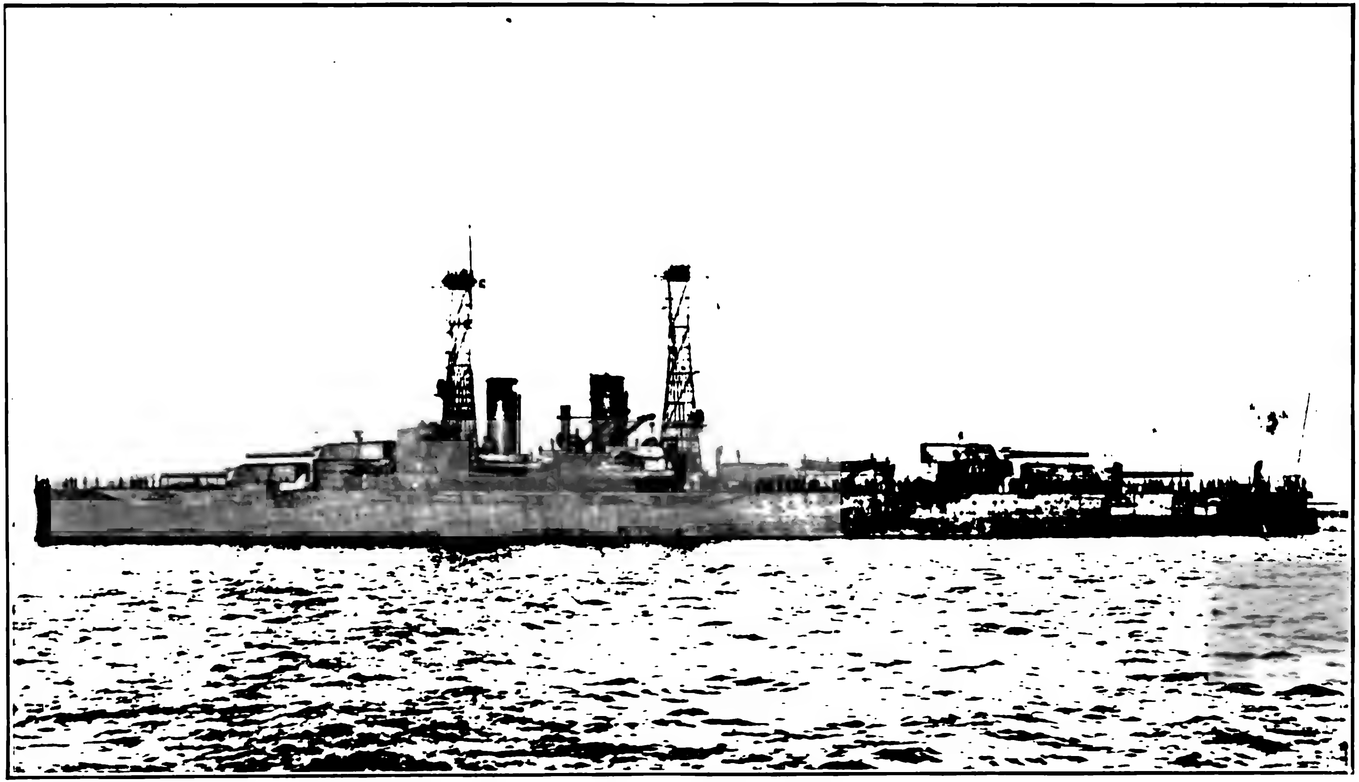
VOLUME XXVII.

WASHINGTON, D. C.

R. BERESFORD, PRINTER, 605 F STREET, N. W.

1915.

INDEX TO VOLUME XXVII, 1915.



U. S. S. "NEW YORK."

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXVII. FEBRUARY, 1915. No. 1.

U. S. S. *NEW YORK*.

DESCRIPTION AND OFFICIAL TRIALS.

BY HENDERSON B. GREGORY, ASSOCIATE.

The *New York* is a twin-screw battleship fitted with reciprocating engines, and designed for a speed of 21 knots, at 27,000 tons displacement, with the main engines developing 28,100 I.H.P. She is also known as Battleship No. 34, and was built at the Navy Yard, Brooklyn, N. Y., in compliance with an Act of Congress approved June 24, 1910, authorizing the construction of two first-class battleships, and which required that one be built at a navy yard. Her sister ship is the *Texas*, built by the Newport News Shipbuilding and Dry Dock Co., of Newport News, Va., whose principal features and trial data are described in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. xxvi, No. 1.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	565–00
on L.W.L., feet and inches	565–00
over all, feet and inches	572–07¾
Breadth, extreme, on L.W.L., feet and inches	95–02⅝
molded, feet and inches	94–10⅛
Depth molded, main deck at side M.S., feet and inches	48–08⅛
Draught to L.W.L., feet and inches	28–05⅜
Displacement corresponding, tons	27,000
Ratio of length to beam	5.944
Displacement per inch at L.W.L., tons	92
Area of midship section, square feet	2,652
L.W.L. plane, square feet	38,560
wetted surface, square feet	65,700
Coefficient of fineness, block	0.61
midship section	0.99
L.W.L. plane	0.706

GENERAL DESCRIPTION OF HULL.

The hull is of steel throughout and of typical American design, with all turrets on the center line, two smoke pipes and the usual cage masts.

Masts.—There are two cage masts, with floors 113 feet 1½ inches high above the load water line, located at about frames Nos. 51½ and 76½; on which are located the spotter's tops and searchlight platforms—mounting two lights each; wireless, signal yards, etc.

Navigating Bridge.—The navigating bridge is just above the bridge deck and between the forward mast and conning tower. It extends athwartship to the side of the bridge deck.

Bridge Deck.—This is a small deck at the base of the forward mast, extending from frames Nos. 44½ to 54. It contains the chart house, and its forward end is taken up with the conning-tower foundations. It has portable extensions at either side, reaching to the vessel's sides.

Superstructure Deck.—This deck extends from the after side of turret No. 2 to frame No. 54, and contains a deck house in which are the captain's quarters.

Main Deck.—The main deck, on which is located the main battery, described elsewhere, is a weather deck throughout. Forward are located the windlass and a deck winch, and far aft a capstan. There are four deck houses, as follows: One forward, in which are the admiral's and chief of staff's quarters, the captain's and fleet officers' offices. In another, at the base of the forward smoke pipe, are the officers' galley, bakery, lamp room and lockers. A third, at the base of the after smoke pipe, contains the blacksmith shop, foundry, butcher shop and lockers. The fourth, below the after mast, is exclusively for the crew's galley. The two boat cranes are located amidship, port and starboard.

Gun Deck.—The gun deck extends from the stem to the stern. Forward of the diagonal armor are the wardroom officers' quarters. Within the casemate is part of the 5-inch battery, described elsewhere; printing office, armory, wardroom officers' bath and water closets, firemen's and mess attendants' wash rooms, drying rooms, coaling engines and bread rooms, general mess condiment and issuing room, lucky bag and post office. Aft of the diagonal armor is the crew's space, executive, ordnance and engineer officers' offices, paymaster's office, ship's stores, surgeon's examining room, dispensary, operating room and sick bay, laundry, master-at-arms and sergeant of marines' staterooms, general mess pantry, chief petty officers' and crew's washrooms and water closets, and electric capstan motor.

Half Deck.—This deck is forward of the forward diagonal armor, between the gun and protective decks. On it are located the junior and warrant officers' quarters, starboard and port respectively, and cleaning-gear room in the peak.

Protective and Berth Decks.—The protective deck extends at one level between frames Nos. 18 and 137, sloping off aft from frame No. 122. Forward of frame No. 18 it is dropped abruptly to the upper platform level and carried forward to the bow, the corresponding interval at former level being replaced by the berth deck, on which are located cofferdam, paint

and lamp rooms, commissary stores and the chain lockers. Above the sloping portion of the protective deck from frame No. 122, the berth deck, containing the chief petty officers' and crew's spaces, is carried aft at the protective-deck level. On the protective deck, from forward aft, are located forward ice-machine room, stores, blower rooms, crew's space, central station, band room, storage-battery charging station, coal bunkers, ammunition passages, boiler hatches, evaporator room, blower rooms, prison, clothing and small stores, crew's space, general workshop, ice machine and refrigerating rooms.

Upper Platform.—Next below the protective deck is the upper platform. Forward of the boiler rooms are electrical stores, distribution-board room, sub-central, radio room, intercommunication room, magazines, handling and shell rooms, windlass machinery, stores, pump room, chain lockers, paint and oil rooms and cofferdam. Extending through the boiler rooms, on the center line, is the wiring passage. The forced-draft blower rooms are located at this level in each fireroom, with coal bunkers outboard. Between the boiler and engine rooms is another distribution-board room; magazines, handling and shell rooms, with steam-pipe passages outboard. Abreast of the engine rooms are blower and store rooms. Abaft of the engine rooms are magazines, handling and shell rooms.

Lower Platform.—Like the former, the lower platform is interrupted by the machinery spaces. Forward are the forward trimming tanks, sails and awnings, hold, navigator's and ordnance stores, magazines, handling and shell rooms, torpedo room and forward dynamo room. Coal bunkers extend abreast of the boiler rooms, port and starboard, and the after dynamo room, together with magazines, handling and shell rooms are between the boiler and engine rooms. Again store rooms outboard of the engine rooms; and store rooms, steering-engine room, magazines, handling and shell rooms, pump and steering-gear rooms aft.

Hold.—In the hold, from forward aft, are located the for-

ward trimming tanks, fresh-water tanks, stores, forward dynamo-condenser room, boiler rooms, with coal bunkers outboard, after dynamo-condenser room, provisions, engine rooms, with wiring passage between; engineer's and ordnance stores, and the after trimming tanks.

Double Bottoms.—The double bottoms extend from frames Nos. 9 to 122. That portion under the boiler rooms, frames Nos. 61 to 73, forms the reserve feed-water tanks. Fuel oil is carried in tanks under the machinery spaces, frames Nos. 78 to 99. All other double bottoms are dry.

Capacities of Double-Bottom Compartments.

Com'p't.	Fresh water, tons.	Salt water, tons.	Com'p't.	Fresh water, tons.	Salt water, tons.
A–92	56.16	57.78	B–89	65.36	67.22
A–93	53.52	55.03	B–90	65.36	67.22
A–94	68.22	70.19	B–91	65.23	67.14
A–95	68.22	70.19	B–98	65.31	67.19
A–96	101.91	104.83	B–99	65.31	67.19
A–97	92.86	95.50	C–98	61.11	62.86
A–98	63.83	65.56	C–99	61.11	62.86
A–99	97.11	99.89	D–97	88.08	90.61
B–86	51.36	52.83	D–98	89.47	92.03
B–87	51.41	52.89	D–99	107.16	110.25
B–88	65.36	67.22			

COMPLEMENT.

The ship's complement will be approximately as follows:

Flag Officer	1
Chief of Staff	1
Commanding Officer	1
Wardroom Officers	30
Junior Officers	18
Warrant Officers	12
Crew (including 72 marines)	1,006
Total	1,069

BATTERY.

There are ten 14-inch guns in the main battery, arranged in pairs, in five turrets on the main deck along the center line of the vessel. Turrets Nos. 1 and 2 are forward, the latter being placed at sufficiently high elevation to permit ahead fire over the top of the former. Turret No. 3 is between the after mast and the engine hatches, and Nos. 4 and 5 are grouped abaft the engine hatches, the former elevated for astern fire over the latter.

The turrets are electrically operated, each containing the following electric apparatus:

No.	Character of Apparatus (Diehl Motors).	H.P. each.
2	Training motors	25
2	Upper ammunition hoists............	40
2	Lower ammunition hoists............	10
2	Elevating motors	15
2	Rammers	10

A secondary battery, of twenty-one 5-inch rapid-fire guns for torpedo defense, is also provided. Nineteen of these guns are distributed along the gun deck as follows: Divided port and starboard—four in the officers' quarters forward, ten in the casements and four in the crew's space aft, with one at the extreme stern on the center line. The two remaining guns are mounted on the bridge deck, port and starboard.

There are twenty-five electric-chain ammunition hoists for the secondary battery, each driven by a 3-horsepower Diehl motor.

The following smaller guns are also provided:

4 3-pdr. guns for saluting;
2 1-pdr. guns for boats;
2 3-inch field pieces;
2 0.30-caliber machine guns.

There are also four 5-m. by 21-inch submerged torpedo tubes forward. For charging the torpedoes, two Mark XIV,

Ingersol-Rand, electric-driven air compressors are provided. Each has a capacity of 30 cubic feet of air per minute, at 2,500 pounds pressure per square inch. The driving motors are 52 horsepower each. The plant is provided with two accumulators of 25 cubic feet capacity each.

SMALL BOATS CARRIED.

The following small boats comprise the ship's regular allowance and are carried on the main deck and in skid-deck beams at the superstructure-deck level abeam of the smoke pipes:

2 50-foot steamers;
1 40-foot steamer;
2 50-foot motor sailing launches;
2 40-foot motor sailing launches;
1 31-foot racing cutter;
2 30-foot whale boats;
2 20-foot dinghies;
2 14-foot punts.

There is also a stowage position for a flag officer's barge, which, in case the ship is a division flagship, would be a 40-foot motor barge.

When used as the flagship of the commander-in-chief the following boats are taken on board, replacing selected boats of the regular equipment:

1 50-foot motor barge for commander-in-chief;
1 35-foot motor boat for chief-of-staff.

Two electrically-operated boat cranes are provided for handling the boats, except the whale boats, which are hung in davits aft. Each crane has two operating gears—one for turning and the other for hoisting. The operating gears are driven by Diehl motors of 40 and 50 horsepower, respectively.

ANCHOR WINDLASS.

The anchor engine is located on the upper platform, frames Nos. 18 to 24. The engine is of the horizontal, double-cylinder, reversible type, built by the Hyde Windlass Co., of Bath, Maine. The cylinders are 16 inches diameter each by 16 inches stroke.

The windlass is driven by a worm gearing direct from a worm on the engine crank shaft. Attached to the worm gearing are two vertical shafts, each fitted on its upper end, at the main deck, with a wildcat, and so arranged that the wildcats can be operated together or independently of each other.

DECK WINCH.

An electrically-operated deck winch is located on the main deck, frames Nos. 22–24, center line. It is compound geared and has a lifting capacity of 16,000 pounds at 50 feet per minute, or 4,000 pounds at 250 feet per minute. The winch was made by the American Engineering Co., and is operated by a 35-horsepower Diehl motor.

COALING ENGINES AND GEAR.

There are two steam-driven coaling engines made by the Lidgerwood Mfg. Co. They are located amidship on the gun deck, frames Nos. 64 to 66. The engines are of the vertical, double-cylinder type, with cylinders 10 inches diameter by 10 inches stroke.

Each engine, through miter gears, drives an athwartship shaft, which in turn drives, port and starboard, fore-and-aft shafts; to which are geared eight gypsy heads each, which can be thrown in or out of gear at will by friction clutches. The athwartship shafts are cross-connected so that either engine can operate all gypsy heads if necessary. The capacity of the gear is 2,400 pounds on each gypsy head, at 200 feet per minute.

CAPSTAN.

Located at frames Nos. 128–129, center line, on the main deck, is an electric capstan, made by the American Engineering Co. It is compound geared, of same capacity as the deck winch and operated by a similar motor, located below on the gun deck.

STEERING ENGINE AND GEAR.

The steering-engine room is on the starboard side of the center line, just abaft of the engine rooms, access being from the starboard engine room. The engine, of the American Engineering Co.'s vertical, double-cylinder type, has cylinders 21 inches in diameter by 16 inches stroke.

From the steering engine a shaft is led aft to the tiller room, where is located the main steering gear, consisting of a right-and-left-hand screw, on which are two driving nuts direct connected by side rods to the rudder-stock crosshead. The screw is operated through gearing by the shaft, or emergency hand-steering gear, either of which may be disconnected when not in use.

In addition to the steam-steering engine, an electric gear is installed, driven by a 150-horsepower Diehl motor. It is located in the steering room and operates the steering gear through the engine shaft, suitable clutches being provided for throwing the engine and the motor in or out of gear.

The steering engine is controlled by telemotor, fitted in duplicate, from the steering platform, conning tower and central station. It may also be direct controlled by a handwheel in the steering room aft and one at the engine.

The control of the electric gear is from the same stations, steering-engine room excepted, as the steam gear.

The usual emergency hand gear is located in the steering room. It consists of four large hand-steering wheels mounted on a common shaft, which is connected through a train of gears and suitable cutout clutches, with the steering gear.

LAUNDRY.

A well equipped laundry, with capacity to wash for about 100 men, is located on the gun deck, frames Nos. 106 to 114, port side.

The laundry machinery is driven by a 6-horsepower Diehl electric motor, through suitable shafting and belting, and comprises the following apparatus:

1 32-inch × 54-inch double-geared, split-head, metal washer;
2 Wash tubs;
1 Soap tank (90 gallons);
1 Starch kettle (10 gallons);
1 20-inch Hogan extractor;
1 16-inch flatwork ironer;
1 Steam-heated bosom press;
1 30-inch steam-heated shoe, reverse body ironer;
1 Ironing table;
1 Drying room (conveyor type).

GALLEYS AND BAKERY.

The galleys and bakery are located in deck houses on the main deck, and, except for the customary steam kettles and coffee urns, are electric throughout; no coal ranges nor bake ovens being used. The officers' galley contains a five-section electric range, and that for the crew one of ten sections. The ranges were supplied by the General Electric Co.

Located in the crew's galley are the following electrically operated machines:

No.	Character of Machine.	H.P. of Motor.
1	Potato peeler	1
1	Meat chopper	¾
1	Cake mixer	½
1	Ice cream freezer	2

The bakery is equipped with two electric bake ovens and one dough mixer driven by a 2-horsepower motor.

An electrically-driven dish-washing machine, with ¾-horsepower motor, is provided for the general mess pantry.

DRAINAGE SYSTEM.

Main Drain.—The main drain, 15½ inches inside diameter, runs from the forward fireroom, in a single pipe, port side, to the forward bulkhead of the engine room, where it branches into two full-size pipes—one continuing aft through the port engine room, and the other athwartship, thence aft through the starboard engine room, both connecting to flanges provided on the main circulating-pump suction pipes.

There is a 15½-inch stop-check valve in the system in each engine and boiler compartment, for draining same, operated at place and from the protective deck. There is also a stop-lift-check valve at each connection to the circulating-pump suction pipes, so interlocked with the main injection valves that the latter must be closed before the former can be opened, as a safeguard against flooding the main drain from the sea.

As auxiliary connections to the main drain proper, there are two 5½-inch suction branches, via the secondary drain, to the engine-room fire and bilge pumps. These branches are each fitted with a Macomb strainer and a stop valve at the main.

Secondary Drain.—The secondary drain extends in single line, forward. Where it joins the fireroom system, there is a Macomb strainer and stop valve at junction. It is 4½ inches in diameter, with full-size suction branch from the drain tank forward. and 3-inch branches from the double bottoms and forward trimming tanks.

In the firerooms the main is 5½ inches in diameter, with branches to double bottoms and fireroom bilge wells, 3 inches and 5½ inches in diameter, respectively. Each fireroom fire and bilge pump has a 5-inch suction connection from the main, and a discharge to the sea of same diameter.

Each engine room is provided with a complete secondary drainage system. The main is 5½ inches in diameter, with

two full-size suction connections from the bilge wells. The two engine-room lines have a 5½-inch cross-connection, from which are taken 4½ and 3-inch connections for drainage system aft and the port shaft alley, respectively; the starboard shaft-alley drain being taken off the 4½-inch branch leading aft. The fire and bilge pumps in each engine room have a 7-inch suction from the main, dividing into the two 5-inch connections—one to each pump. The combined overboard discharge is 7 inches in diameter.

In addition to the former, there is a shaft bilge pump located in each shaft alley, with 4½-inch independent suctions from the after engine-room bilge and shaft alley, and overboard discharge of same size.

The after system is a single pipe 4½ inches diameter, with 3-inch branches, to the double bottoms and after trimming tanks. It is fitted with Macomb strainer and stop valve where it joins the engine-room system.

All suction connections from bilge wells in machinery spaces are fitted with Macomb strainers and stop-check valves operated at place only. The double-bottom suctions are fitted with stop-lift-check in way of machinery spaces, and stop-check valves elsewhere; the former being operated at place only, and the latter at place and from the protective deck.

There are two 2½-inch suction connections from the secondary drain—one in the forward fireroom, and the other in the port engine room, for hand pumps.

Chain-Locker Drains.—A 3-inch drain is led from the bottom of the chain locker to the drain tank forward, with stop valve at the tank. A 2-inch deck drain, from the windlass machinery space is also led into this pipe.

Torpedo-Tube Drains.—There is a 4½-inch drain pipe from each torpedo tube, combining into a common pipe of same diameter, fitted with stop valve and led to the drain tank forward.

Turret and Deck Drains.—2-inch deck drains are installed as required, each fitted with a valve on deck. Those forward

unite into a common main of 3 and 4½ inches diameter, with stop valve at end, discharging into the forward fireroom bilge. Amidship they form a common 3½-inch pipe, with stop valve at end, discharging into the port engine-room bilge. Aft the combined line is 3, 4 and 4½ inches diameter, discharging through stop valve into the port engine-room bilge.

TRIMMING TANKS.

There are four trimming tanks—two forward and two aft, of the following capacities:

Com'p't	Fresh water, tons.	Salt water, tons.	Com'p't	Fresh water, tons.	Salt water, tons.
A–1	90.66	93.25	D–12	191.72	197.20
A–2	117.84	121.21	D–13	48.50	49.89

The tanks are flooded from the sea by a 6-inch sea chest, forward and aft, and are pumped out through the secondary drain.

FIRE MAIN.

The fire main is supplied by eight fire and bilge pumps, located in the engine and firerooms. The two distiller circulating pumps can also be used on the fire main in emergencies.

The main is entirely below the protective deck, and extends throughout the machinery spaces and forward to frame No. 42, in two 6-inch lines, port and starboard. It is cross-connected at frames Nos. 42, 89 and 104, and supplied by two 6-inch risers from the engine-room fire and bilge pumps, and four 5-inch risers from the fireroom fire and bilge pumps, with cutout valves at the main.

Forward of frame No. 42 the main extends in single line to frame No. 18, in size 6 and 4 inches diameter.

Aft of the engine rooms a single line, 6, 5 and 4 inches in diameter, leads astern to frame No. 125, where it joins the independent sanitary system for the crew's water closets and

washroom aft. There is a stop valve where it joins the latter.

On the protective deck at frames Nos. 41 and 104, starboard, there are bypasses to the sanitary system. The bypasses are 6 inches in diameter and fitted with locked valves.

At frame No. 77, protective deck, amidship, there are two 4½-inch emergency connections for supplying circulating water to the distillers, which are also used for supplying the fire main by the distiller circulating pumps.

There are 6-inch flooding connections to all coal bunkers, branches to magazines flooding and sprinkling, and risers as required, leading to fire plugs on the decks above, distributed as follows:

Lower platform	1	Gun deck..............	25
Upper platform	14	Main deck	15
Protective deck	11	Superstructure deck	2
Berth deck	1		—
Half deck.............	3	Total...............	72

SANITARY AND FLUSHING SYSTEM.

The flushing main is 6 inches in diameter, and is supplied by six 5-inch risers from the fire and bilge pumps, and two 6-inch bypasses, one forward and one aft, from the fire main.

It is carried in the protective-deck space, close under the gun-deck beams, and extends from frames Nos. 41 to 104, at which point it reduces in size to 4 inches and continues aft to frame No. 121, where it rises to the gun deck and leads aft to frame No. 129, joining the crew's independent flushing main, with stop valve at junction.

Branches, as required, are led to the chief petty officers' washroom and water closets, sick-bay, bath, laundry, general mess pantry, galleys, bakery, firemen's washrooms, junior, warrant and wardroom officers' lavatories and water closets, etc.

Aft there is an independent system for the exclusive use of the crew's washroom and water closets. This system is supplied by two motor-driven, direct-connected, centrifugal pumps*, of about 500 gallons per minute capacity each. The main is 5 inches in diameter, with the necessary branches to the plumbing fixtures.

FRESH-WATER SYSTEM.

The following fresh-water tanks are provided:

Compartment or tank.	Location.	Between frames.	Gallons.
A-6	Hold-port	18-21	9,760
A-7	" starbd.	18-21	9,760
A-8	" port	21-24	11,860
A-9	" starbd.	21-24	11,860
Main gravity tank	Top of deck house	59-62	2,000
Aux. gravity tank	Top of chart house	52-53	150
Firemen's supply tank	Top of deck house, port	64-65	450
Do.	Do. stbd.	61-62	450
Battle dressing-station supply tank	Protective deck, stbd	48-49	125
Do.	Do.	114-115	125

The hold tanks have two 2½-inch filling connections from the ship's sides forward, and also a 3-inch connection from the distiller main. From these tanks water is pumped, through the fresh-water main, to the main gravity tank, by means of two triple-plunger electric-driven pumps.* There is also a 1½-inch discharge connection from these pumps for filling the auxiliary gravity tank, which supplies water, through an independent system, to the officers' quarters above the main deck.

The main gravity tank supplies the entire fresh-water system, except as noted in preceding paragraph, including the various supply tanks. In case the gravity tank is out of order, all parts of the ship supplied by it may be supplied by the pumps direct through the main. The main is 3 inches in diameter, and has a 3-inch branch to the main gravity tank and

*See Table II.

other branches as required to the supply tanks, officers' baths, pantries, galleys, bakery, sick bay, crew's washroom, scuttle-butts, etc.

MAGAZINE FLOODING AND SPRINKLING.

There are three flooding systems for the three groups of magazines—forward, amidship and aft. Each system floods the magazines of its group on the lower platform, while magazines on the upper platform are flooded or sprinkled from the fire main.

Each system is provided with a 9-inch sea chest, from which pipes are led, with branches to the various magazines as required. Adjacent to the sea connections are stop valves operated at place and from the protective deck.

A sprinkling system is fitted in all magazines. It is supplied by the fire main, and in the magazines consists of 2½-inch brass pipe perforated on the under side, so that each powder tank can be sprinkled. There are cutout valves for each magazine operated at place and from the protective deck.

MAGAZINE COOLING SYSTEM.

Provision is made for cooling, by brine circulation, all magazines, except those containing saluting powder and small-arms ammunition. For this purpose 1-inch galvanized-iron pipe coils are fitted overhead in the magazines that require cooling, through which the previously cooled brine is circulated. Suitable drip pans are fitted under all coils to catch the condensation.

REFRIGERATING PLANT.

The refrigerating apparatus is of the CO_2 type, as manufactured by the Kroeschell Bros. Ice Machinery Co. It consists of five vertical and one horizontal electric-driven compressors, 2½-inch bore by 8-inch stroke, direct connected to electric motors—15-horsepower General Electric units designed to run at 100 revolutions per minute. Each compressor

is capable of producing the cooling effect of 6 tons of ice per day.

The installation is divided into two independent plants—one forward, on the protective deck, frames Nos. 18 to 21 port, for the exclusive use of the forward magazines. It consists of one horizontal compressor set, together with its water circulating and brine pumps,* brine tank, condenser and the necessary piping and fittings. The other, or main plant, is located in the refrigerating-machinery room on the protective deck, port of the engine hatches. There are five vertical compressor sets in this group, arranged so that any or all machines can be used on the refrigerating rooms, ice-making tank, scuttle butts or magazines, including those forward. There is one large condenser for this group of compressors, together with two electric-driven centrifugal water circulating pumps,* three electric-driven centrifugal brine circulating pumps,* a brine cooler, ice-making tank, and all the necessary piping and fittings. The refrigerating rooms, ice-making tank, scuttle butts and magazines are brine cooled; the brine, previously cooled in the brine coolers, being forced through the system by the brine circulating pumps and returned to the coolers, from which the cycle is repeated. The brine coolers are cooled by the CO_2 gas, which is taken from the condenser liquid receivers, circulated through the brine coolers and returned to the compressors.

There are four refrigerating rooms, on the protective deck, just abaft of the main refrigerating machinery room. All the rooms are fitted with 1-inch galvanized iron-pipe coils, through which the cooling agent is circulated. The crew's compartment serves as vestibule for the butter and meat rooms, and the officers' cooling room has no vestibule, its door opening direct to the passage. A hatch is provided through the gun deck for charging the meat room. The rooms are cork insulated in the usual manner.

*See *Table II.*

RESERVE-FEED TANKS AND FILLING CONNECTIONS.

The following double bottom compartments are fitted up as reserve-feed tanks:

Comp't.	Fresh water, tons.	Comp't.	Fresh water, tons.
B–92	53.78	B–95	53.69
B–93	53.78	B–96	53.78
B–94	53.78	B–97	53.69

A 3½-inch filling pipe is led across the ship on the gun deck, frame No. 64, both outboard ends of which are carried through the main deck and fitted with two 2½-hose valves each. Amidship it has a 3-inch by-pass filling connection to the fresh-water main, fitted with locked valve, and a 4-inch branch led down into fireroom No. 3, where it joins the filling and suction manifold, through which the various compartments are filled.

FUEL-OIL TANKS AND FILLING CONNECTIONS.

The double bottoms under the machinery spaces, frames Nos. 78 to 99, are arranged for carrying fuel oil as follows:

Compt.	Gallons.	Compt.	Gallons.
C–90	20,942	C–94	17,000
C–91	20,942	C–95	17,000
C–92	17,575	C–96	15,233
C–93	17,563	C–97	14,947

For filling the fuel-oil compartments, two 6-inch pipes are led across the ship under the main deck, with hose valves at the ship's side, frames Nos. 100–101 starboard and 103–104 port; and on the main deck at frames Nos. 85 starboard and 89 port. These pipes are combined into an 8-inch pipe, which leads down the starboard engine hatch to the manifolds in the engine rooms, individual oil compartments being filled through the manifolds and suction pipes from the tanks.

An 8-inch overflow pipe is fitted to the filling pipe in engine hatch, discharging overboard.

COAL-BUNKER CAPACITIES.

Comp't.	Tons.	Comp't.	Tons.
B-5-P	91.5	B-102	81.7
B-5-S	90.6	B-103	81.9
B-7-P	145.8	B-104	84.4
B-7-S	145.8	B-105	84.4
B-9-P	163.3	B-106	80.5
B-9-S	163.3	B-107	80.6
B-11-P	151.0	B-108	78.9
B-11-S	151.0	B-109	78.9
B-13-P	135.1	B-110	97.3
B-13-S	135.1	B-111	97.3
B-15-P	134.3	B-112	96.6
B-15-S	134.3	B-113	96.6
B-17-P	105.7		
B-17-S	105.7	Total	2,892.5

VENTILATING SYSTEM.

Artificial ventilation is provided where necessary for all quarters, living spaces, passages, storerooms, magazines, engine rooms, dynamo rooms, evaporator room, etc. There are forty-nine ventilating fans and motors, each on its own circuit. In general air is supplied on the plenum system to the different compartments requiring ventilation. The toilet spaces and engine rooms are also provided with the exhaust system. The ventilating fans were made by the B. F. Sturtevant Co., Boston, Mass., and the motors by the Diehl Mfg. Co., except those for the engine-room exhaust fans, sets Nos. 45 to 48 inclusive, which were built by the General Electric Co.

Table I contains the particulars of the ventilating fans and motors.

Heater boxes are fitted in the ventilating ducts to all quarters below the main deck, crew's space, etc., for heating the incoming air in cold weather; no other heating apparatus being provided for these spaces.

TABLE I - VENTILATING FANS - U.S.S. NEW YORK.								
SET No.	LOCATION			MOTOR			FAN	
	DECK	P OR S	FRAMES	OPEN OR ENCLOSED	HP	R.P.M.	SUPPLY OR EXHAUST	CAPACITY CU. FT. OF AIR PER MIN.
1	TURRET I	S	26-27	O	4.8	790	S	5000
2	" I	P	" "	O	"	"	S	"
3	" II	S	36-37	O	"	"	S	"
4	" II	P	" "	O	"	"	S	"
5	" III	S	88-89	O	"	"	S	"
6	" III	P	" "	O	"	"	S	"
7	" IV	S	109-110	O	"	"	S	"
8	" IV	P	" "	O	"	"	S	"
9	" V	S	119-120	O	"	"	S	"
10	" V	P	" "	O	"	"	S	"
11	GUN	C.L.	100-101	E	2.19	1055	E	2500
12	"	P	113-114	E	0.88	1560	S	1000
13	"	P	124-125	O	2.19	1055	S	2500
14	HALF	C.L.	12-13	E	"	"	E	"
15	BERTH	C.L.	" "	E	0.88	1560	S	1000
16	"	S	18-19	O	2.19	1055	S	2500
17	"	S	30-31	O	7	590	S	8000
18	"	P	32-33	O	4.37	745	S	5000
19	"	S	" "	O	"	"	S	"
20	"	P	35-36	O	"	"	S	"
21	"	P	45-46	O	3.5	835	S	4000
22	"	P	46-47	O	"	"	S	"
23	"	S	" "	O	"	"	S	"
24	"	P	48-49	O	8.75	500	S	10000
25	"	S	" "	O	"	"	S	"
26	"	P	76-78	O	7	590	S	8000
27	"	S	" "	O	7	"	S	"
28	"	P	81-82	E	0.88	1560	E	1000
29	"	S	" "	E	"	"	E	"
30	"	P	83-84	O	2.19	1055	S	2500
31	"	S	" "	O	"	"	S	"
32	"	S	" "	O	3.5	835	S	4000
33	"	S	86-87	O	7	590	S	8000
34	"	P	92-93	O	7	"	S	"
35	"	S	" "	O	7	"	S	"
36	"	P	114	O	8.75	500	S	10000
37	"	S	"	O	7	590	S	8000
38	UPPER PLAT.	C.L.	42-43	E	0.25	2200	E	425
39	" "	C.L.	137-138	O	3.5	835	E	4000
40	" "	P	91-92	E	10.5	485	S	12000
41	" "	S	" "	E	"	"	S	"
42	" "	P	93-94	E	"	"	S	"
43	" "	S	" "	E	"	"	S	"
44	STR'G. FLAT	P	123-124	O	2.19	1055	S	2500
45	BERTH	S	94-95	O	8.5	485	E	12000
46	"	P	" "	O	"	"	E	"
47	"	S	95-96	O	"	"	E	"
48	"	P	" "	O	"	"	E	"
49	MAIN	S	48-49	E	0.25	2050	S	425

HEATING SYSTEM.

All staterooms and quarters below the main deck, crew's space, etc., are heated by the thermo-tank ventilating system; heater boxes provided with suitable steam coils being placed in

the air ducts for heating the air supplied to these compartments. The remaining portions of the vessel, *i.e.*, quarters on the main, superstructure and bridge decks, are heated by the customary pipe-coil steam radiators.

The heating system is divided into two main sections, one forward and one aft. The forward section is taken off the main steam cross-connection pipe in fireroom No. 2, starboard side, with stop and reducing valves adjacent to the steam line, and is subdivided into five independent circuits as follows:

Circuit No.	Description.	Working pressure. Pounds.
1	Crew's and officers' galleys and bakery......	20
2	Bath, shower and lavatory-water heaters forward of frame No. 74; pantries and fresh-water tanks	20
3	Heater boxes (2), protective deck, frames 32–33 port and 34–35 starboard..........	50
4	Heater boxes (4), protective deck, frames 46–47 port, 45–46 starboard and 51–52 port and starboard	50
5	Radiators in quarters on main, superstructure and bridge decks, and fresh-water tanks on chart house	50

The after section is taken off the auxiliary steam loop in engine rooms, with stop and reducing valves at the line, and is subdivided into eight independent circuits as given below:

6	Heater boxes (2), gun deck, frames 92–93 port and starboard.....................	50
7	Heater boxes (2), protective deck, frames 91–92 port and starboard.................	50
8	Heater box in steering room...............	50
9	Heater boxes (3), gun deck, frames 124–125 center line, protective deck, frames 112–113 port and starboard.....................	50
10	Heater boxes (2), gun deck, frames 112–113 port and starboard.....................	50

11	Heater box for hospital space, gun deck, frames 111–112, starboard, and radiator in operating room	50
12	Laundry and general mess pantry	20
13	Bath, shower and lavatory heaters aft of frame No. 74	20

The drains, from circuits Nos. 3, 4 and 5, are collected into a manifold and led to a common trap in fireroom No. 2. Drains from circuit No. 1 and pantries and fresh-water tank coils, circuit No. 2, are led to a trap located in the same fireroom. The drains from water heaters, circuits Nos. 2 and 13, are led direct to the auxiliary exhaust line. Circuits Nos. 6, 7, 8, 9, 10 and 11 are led to a drain manifold in the port engine room, which connects to a trap in the same engine room. There is also an independent trap in the port engine room for the drain from circuit No. 12. All traps discharge into the low-pressure trap discharge main.

The following is a list of the heater boxes in the ventilating ducts:

No.	Location.			Heating surface, sq. ft. each.	Steam from circuit No.	On ventilating system No.
	Deck.	Frames.	P. or S.			
2	Gun.	92–93	P. and S.	186.4	6	34 and 35
1	"	111–112	S.	58.3	11	36
2	"	112–113	P. and S.	129.7	10	36
1	"	124–125	C. L.	174.3	9	13
1	Protective.	32–33	P.	355.1	3	18
1	"	34–35	S.	355.1	3	19
1	"	46–47	P.	161.1	4	24
1	"	45–46	S.	124.8	4	25
2	"	51–52	P. and S.	259.1	4	24 and 25
2	"	91–92	P. and S.	115.6	7	34 and 35
1	"	112–113	P.	69.4	9	36
1	"	112–113	S.	82.8	9	36
1	Steering-engine room.	125	P.	80.3	8	....

MAIN ENGINES.

There are two main engines designed to develop collectively 28,100 indicated horsepower, when making 125 revolutions

per minute. They are placed abreast in two separate watertight compartments, as shown in Plate I.

The engines are of the vertical, inverted-cylinder, direct-acting, four-cylinder, triple-expansion type, turning outboard when going ahead. The order of the cylinders, beginning forward, is forward low-pressure, high-pressure, intermediate-pressure and after low-pressure. All crank angles are 90 degrees, the sequence of cranks being high-pressure, intermediate-pressure, forward low-pressure, and after low-pressure.

Bedplates.—The bedplates are of cast steel, in three sections each, bolted together and supported on keelson plates. Proper seatings and facings are provided for the main bearings, columns, etc.

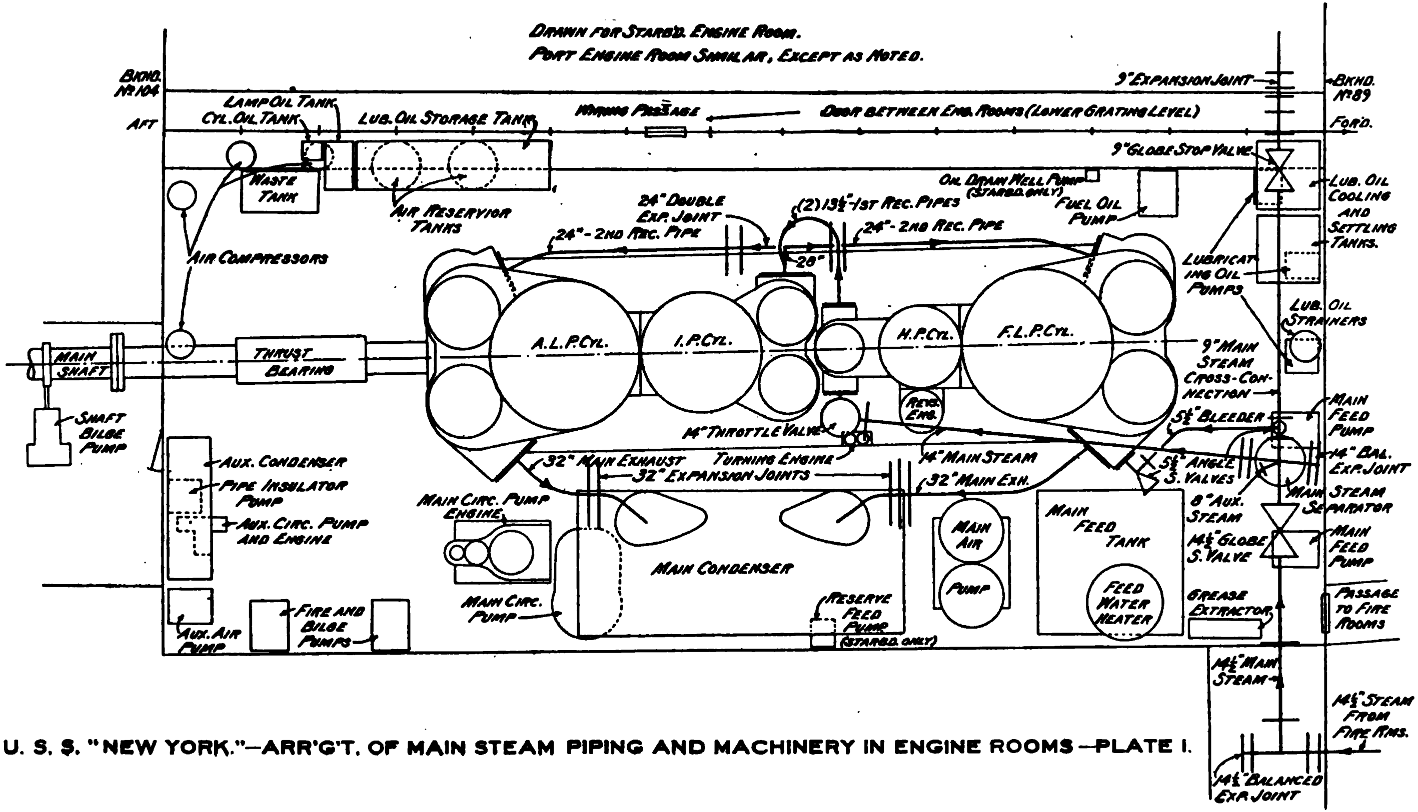
Main Bearings.—The main bearings consist of a lower brass and cast-steel cap, each lined with white metal and cored for the circulation of cooling water.

Framing.—The engine frames are of the usual Navy type forged-steel columns, bolted to the bedplate and cylinders, and braced by suitable diagonal, cross and longitudinal stays.

Cylinders.—The cylinders and valve chests are of cast iron, fitted with working liners of close-grained cast iron as hard as can be properly worked. All cylinders, except the high-pressure, are steam jacketed around the working liners and at both ends.

Pistons.—All pistons are of conical design, those for the high-pressure cylinders being of cast iron, all others are of cast steel. The high-pressure followers are of cast iron, and those for the intermediate and low-pressure pistons are forged steel. The high and intermediate-pressure pistons have one solid packing ring each, and each low-pressure piston two rings, cut obliquely into eight sections each, and fitted with brass tongue pieces and lugs. All packing rings are of cast iron and floated by springs.

Piston Rods.—Each piston rod is tapered to fit its piston and secured by a locked nut. The lower end is fitted to a



U. S. S. "NEW YORK."—ARR'G'T. OF MAIN STEAM PIPING AND MACHINERY IN ENGINE ROOMS—PLATE I.

forged-steel crosshead, to which is bolted a cast-steel slipper, white-metal lined. The piston rods are forged steel.

Crosshead Guides.—The go-ahead guides are of cast iron, hollowed for the circulation of cooling water. The backing guides are of cast steel and securely bolted to flanges on the go-ahead guides. The guides are supported by bolting to facings on the cylinders at the upper end, and to a cast-steel girder of "I" section at the lower end; the girder being secured to the inboard engine columns.

Connecting Rods.—The connecting rods are of forged steel, forked at the top to span the crosshead and carrying the cross-head brasses, and "T" headed at the bottom to receive the crank-pin brasses.

Valve Gear.—The engines are equipped with the Stephenson, double-bar link valve gear, fitted with Lovekin assistant cylinders. Piston valves are used throughout; there being one for each high-pressure and two for each intermediate and low-pressure cylinders.

Reversing Gear.—Each main engine is provided with a reversing engine of the vertical, direct-acting type, bolted to the high-pressure cylinder and connected through connecting rods to the reversing-shaft arms; the shaft in turn connecting to the main links by arms and suspension rods. Each reversing engine has a steam cylinder 17 inches in diameter by 22 inches stroke, and an 8½-inch oil-controlling cylinder of same stroke, for taking up shock and for hand operation, a small hand pump being provided for that purpose. The gear is controlled by a floating lever operated at the working platform.

Turning Gear.—The customary turning gear is fitted on each engine, consisting of a double engine, with cylinders 7 inches in diameter by 5 inches stroke. The engines drive, by worm gearing, a second worm, which may be made at will to mesh with a worm wheel fitted on the crank shaft. The turning engines are reversible.

Each turning-engine shaft is also fitted for turning by hand.

Lubricating Gear.—All working and moving parts of the

main engines, except the valve links and valve-stem guides, which are efficiently lubricated by combination sight and wick-feed oil-distributing boxes, located on the main cylinders, are lubricated by the forced-lubrication system, described elsewhere. The crosshead guides are provided with both gravity and forced lubrication.

Water Service.—Water service is provided for each main engine by a 4-inch pipe from the discharge pipe of the main circulating pump. This pipe has suitable branches to the various parts of the main engine, thrust and spring bearings; the discharge being returned to the suction side of the same pump.

Working Platform.—Owing to the fact that the forced-lubrication system requires the crank pits to be totally inclosed by a thin galvanized sheet-steel casing extending up nearly to the bottoms of the cylinders, the working platforms are on the first grating above the engine-room floor, there being a door through the center-line bulkhead, at this level, between the engine rooms.

Main Engine Data.

Working pressure at H.P. chest, pounds gage	265
Revolutions per minute, designed	125
Indicated horsepower, total designed	28,100
Diameter of H.P. cylinder, inches	39
I.P. cylinder, inches	63
L.P. cylinders (2) inches	83
Stroke, inches	48
Ratio, I.P. to H.P.	2.61
L.P. to H.P.	9.06
L.P. to I.P.	3.47
Total expansions	10.92

	Per cent. of volume.		Linear, inches.	
Cylinder clearances:	Top.	Bottom.	Top.	Bottom.
Starboard, H.P.	14.29	13.54	7/16	9/16
I.P.	12.18	14.88	9/16	37/64
F.L.P.	14.06	16.5	9/16	9/16
A.L.P.	13.43	16.1	1/2	35/64
Port, H.P.	14.29	13.54	15/32	5/8
I.P.	12.18	14.88	31/64	19/32
F.L.P.	14.06	16.5	29/64	31/64
A.L.P.	13.43	16.1	15/32	41/64

Valves and valve settings:	H.P.		I.P.		L.P. (each).	
Number and type of valves	1 piston.		2 piston		1 piston.	
Diameter of valves, inches	22$\frac{1}{16}$ top. 22 bottom.		24$\frac{1}{16}$ top. 24 bottom.		34$\frac{1}{8}$ top. 34$\frac{1}{16}$ bottom.	
Travel of valves, inches	10		11		12	
Inside or outside steam	Inside		Outside.		Outside.	
	Top.	Bot.	Top.	Bot.	Top.	Bot.
Width of port, inches	3$\frac{1}{2}$	3$\frac{1}{2}$	3$\frac{1}{2}$	3$\frac{1}{2}$	4$\frac{1}{2}$	4$\frac{1}{2}$
Steam opening, linear, inches	3$\frac{3}{16}$	3$\frac{5}{16}$	3$\frac{5}{16}$	3$\frac{7}{16}$	2$\frac{13}{16}$	3
Exhaust opening, linear, inches.	Full port.		Full port.		Full port.	
Steam lap, inches	1$\frac{13}{16}$	1$\frac{11}{16}$	2$\frac{3}{16}$	2$\frac{1}{16}$	3$\frac{3}{16}$	3
Exhaust lap, inches	−$\frac{7}{16}$	$\frac{1}{4}$	−$\frac{9}{16}$	$\frac{1}{4}$	−$\frac{5}{8}$	$\frac{1}{4}$
Steam lead, linear, inches	$\frac{11}{16}$	$\frac{15}{16}$	$\frac{7}{8}$	1	1$\frac{1}{16}$	1$\frac{1}{4}$
Cutoff, decimal of stroke	85.15	78.65	81.5	73.7	67.06	56.9
mean	81.9		77.6		61.98	

Piston rods, diameter, outside, inches 08¼
axial hole, inches........1½ H.P. and I.P., 4 L.P.
Connecting rods, diameter, crosshead end, inches............... 07½
crank end, inches.................... 09¼
axial hole, inches....1½ H.P. and I.P., 4 L.P.
length between centers, inches................. 96
crank ratio 4
Diameter of throttle valve, inches............................ 14
1st receiver pipes (2), inches.................... 13½
2d receiver pipe, inches............................ 24
main exhaust pipes (2), inches.................... 32
live-steam pipes to receivers, inches................ 03

SHAFTING AND BEARINGS.

There are two lines of shafting, each consisting of a crank shaft in two sections, a thrust shaft, one piece of line shafting, a stern-tube shaft and a propeller shaft, all supported by suitable bearings.

Crank Shafts.—The crank shafts are solid-forged, hollow shafts, in two sections each; bolted together by disc couplings, and carried in seven main bearings. The forward section includes the H.P. and F.L.P. cranks, which are opposite, as are also the I.P. and A.L.P., on the after section. All crank angles are ninety degrees, the two sections being connected with the H.P. crank leading followed by the I.P. crank.

Thrust Shafts.—The thrust shafts are fitted to the usual thrust bearings of the adjustable horse-shoe type, with steady

bearings at each end for supporting the shafts. The shafts are hollow with thrust collars and coupling discs forged integral with the shafts.

Line Shafts.—These shafts are hollow forged, with disc couplings at either end, and each is carried by one steady bearing.

Stern-Tube Shafts.—The stern-tube shafts are covered with a composition casing within the stern tubes and at bearings. They are hollow forged, secured to the line shafts at inboard end by special disc couplings, and to the propeller shafts at outboard end by the usual sleeve couplings, the shafts being taper turned to suit. Each stern tube is provided with two lignum vitae lined bearings for supporting the shafts. There is also one spring bearing in each shaft alley supporting the overhanging ends of these shafts.

Propeller Shafts.—Each propeller shaft is carried by two lignum vitae lined bearings, fitted to the forward and after struts. The shafts are hollow forged, taper turned at both ends to suit the propeller hub and sleeve coupling, and composition bushed at the bearings.

Inboard Coupling.—The inboard coupling consists of a sleeve secured to the stern-tube shaft by four keys. Back of the sleeve is a collar made in halves and secured to the sleeve and to the coupling disc on the line shaft by fitted bolts.

Outboard Coupling.—The outboard coupling is of the solid-sleeve type, taper bored to fit the shafts, and secured to each shaft by two feather keys and one cross key.

Shaft Data.

Crank shafts, length, forward section, feet and inches	**17–03¼**
after section, feet and inches	**19–04**
diameter, inches	**18¾**
axial hole, forward section, inches	**11½**
after section, inches	**10½**
number of cranks	**4**
throw of cranks, inches	**24**
crank angles, degrees	**90**

Crank pins, length, inches	23
diameter, inches	19½
axial hole, forward section, inches	12
after section, inches	10
Thrust shafts, length, feet and inches	16–04¼
diameter, inches	17¾
at bearings, inches	18
axial hole, inches	11½
collars, number	14
thickness, inches	02
space between inches	04
outside diameter, inches	27½
inside diameter, inches	18
bearing surface, square inches	4,752.85
Line shafts, length, feet and inches	18–00
diameter, inches	18
axial hole, inches	11½
Stern-tube shafts, length, feet and inches	51–08⅝
diameter, inches	18½
axial hole, inches	11
Propeller shafts, length, feet and inches	55–09¾
diameter, inches	18½
axial hole, inches	11
Coupling discs, diameter, inches	33
thickness, inches	04
Inboard coupling, diameter of sleeve outside, inches	33
inside, inches	20½
length of sleeve, inches	12
thickness of collars, inches	4
Coupling bolts, number each coupling	8
diameter (taper)* at face of coupling, inches	03⅞
Outboard couplings, length of sleeve, inches	73⅞
diameter of sleeve, inches	24¼

Bearing Data.

Main bearings (white-metal lined):	
Number each engine	7
Diameter, inches	18¾
Length, inches ... two of 15, three of 24 and two of	27¾
Thrust bearings (white-metal lined bearings and shoes):	
Steady bearings, number each	2
diameter, inches	18
length, inches	19

*Parallel bolts for inboard coupling.

Thrust shoes, number each	14
effective surface, square inches	3,046
Line-shaft bearings (white-metal lined):	
Number each engine	2
Diameter, inches	18
Length, inches	26
Stern-tube bearings (lignum vitae lined):	
Number each engine	2
Forward bearing, diameter, inches	$20\frac{5}{8}$
length, inches	48
After bearings, diameter, inches	$20\frac{5}{8}$
length, inches	65
Strut bearings (lignum vitae lined):	
Number each engine	2
Forward bearing, diameter, inches	$20\frac{11}{16}$
length, inches	38
After bearing, diameter, inches	$20\frac{3}{4}$
length, inches	63

PROPELLERS.

There are two twin-screw, three-bladed, propellers of the adjustable-pitch, detached-blade type. They are of manganese-bronze and the blades are secured to the hubs by seven $4\frac{3}{4}$ tap bolts each. The blades are machined true to pitch and the hubs have a taper fit on the shafts, and are secured by a key and nut.

Propeller Data.

Diameter of propeller, feet and inches	$18–08\frac{1}{4}$
hub, feet and inches	4–06
Pitch as set, feet and inches	$19–11\frac{9}{16}$
adjustable from, feet and inches	$18–11\frac{1}{4}$ to $21–02\frac{7}{32}$
Ratio of diameter to pitch	0.9361
Area, projected, square feet	83.43
helicoidal, square feet	98.5
disc, square feet	274.27
Ratio, projected to disc area	0.309
helicoidal to disc area	0.361
Heighth of lower tip of blade above keel, inches	$7\frac{1}{2}$
Immersion of upper tip of blade, inches	$117\frac{3}{8}$

MAIN CONDENSING APPARATUS.

Main Condensers.—There is one main condenser of cylindrical form for each main engine, with tubes rolled into the tube sheets at one end and gland packed at the other. The principal dimensions follow:

Inside diameter, feet and inches	7–09
Thickness of shell (steel), inch	00 7/8
Length betwen tube sheets, feet and inches	15–00
Thickness of tube sheets, inches	1 3/8
Tubes, number	5,340
diameter, outside, inch	00 5/8
thickness, inch	00.065
Cooling surface, square feet	13,104
Diameter of main exhaust nozzles (2), inches	32
auxiliary exhaust nozzle, inches	08
air-pump suction, inches	12
circulating-water inlet and outlet, inches	23

**Main Air Pumps.*—Each main condenser is provided with a Blake, vertical, twin, bucket, single-acting air pump, with steam and water cylinders 14 and 35 inches diameter, respectively, by a common stroke of 21 inches. The suction nozzle is 12 inches and the discharge nozzle 10 inches in diameter.

**Main Circulating Pumps and Engines.*—There is one double-inlet centrifugal circulating pump for each main condenser, driven by a vertical compound engine. The engine is provided with a self-contained forced-lubrication system, with branch connection from the main engine system. The principal dimensions of the pump and engine are as follows:

Capacity of pump, gallons per minute	21,000
Diameter of suction nozzle (2), inches	17
discharge nozzle, inches	23
impeller, inches	42
H.P. cylinder, inches	10
L.P. cylinder, inches	20
Stroke, inches	10
Revolutions per minute	235

*See Table II.

Feed and Filter Tank.—A feed and filter tank of 3,920 gallons capacity is located in each engine room, outboard side and forward of the main condenser. The filter chamber is in the top of the tank and has a capacity of about 814 gallons. The filter has an inner bottom of loose perforated plates and is divided into three compartments, in which is placed the filtering material, by vertical division plates. These partitions are so arranged that the water in passing through the filter will flow under and over in succession, thus assuring the filtering material being always submerged.

Each tank is provided with the following connections:

1 10-inch main air-pump discharge;
1 8-inch cross-connection;
2 5½-inch main feed-pump suctions;
1 4-inch auxiliary air-pump discharge;
1 2½-inch reserve-feed pump discharge (starbd. only);
1 8-inch overflow;
3- and 2-inch vapor pipes (combined into one 3-inch pipe).

ENGINE-ROOM AUXILIARIES.

Auxiliary Condensers.—In each engine room there is an auxiliary condenser connected through the auxiliary exhaust pipe to all the auxiliary machinery. The tubes are rolled into the tube sheets at one end and packed at the other.

They are of the following principal dimensions:

Inside diameter, feet and inches	2–00
Thickness of shell (steel), inch	00$\frac{5}{16}$
Length between tube sheets, feet and inches	6–06¼
Thickness of tube sheets, inch	01
Tubes, number	333
diameter, outside, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet	355.24
Diameter of auxiliary exhaust nozzle, inches	06
air-pump suction, inches	06
circulating-water inlet and outlet, inches	05

**Auxiliary Air Pumps.*—A Blake, vertical, double-acting, single, featherweight, air pump is provided for each auxiliary condenser, with steam and water cylinders of 7½ and 14 inches diameter, respectively, by a common stroke of 12 inches. The suction nozzle is 6 and the discharge nozzle 4 inches in diameter.

**Auxiliary Circulating Pumps.*—Each auxiliary condenser is equipped with a double-inlet centrifugal circulating pump, driven by a vertical, single engine, fitted with a self-contained system of forced lubrication. The general dimensions of the pump and engine follow:

Diameter of suction nozzle, inches	05
discharge nozzle, inches	05
impeller, inches	28
steam cylinder, inches	05
Stroke, inches	04
Revolutions per minute	400

Feed-Water Heater.—Two Reilly multicoil feed-water heaters are installed—one in each engine room. They have 319.8 square feet of heating surface each, and are connected to the main feed lines only. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each connection to the main and auxiliary condensers, opening toward the condenser.

**Main Feed Pumps.*—Two Blake, vertical, double-acting, single, main feed pumps are installed on the forward bulkhead of each engine room. The pumps have independent suctions from the main feed tanks in same engine room and discharge to the boilers through the feed-water heaters or by-passing same.

**Reserve Feed Pump.*—A small Blake, vertical, double-acting, single, reserve feed pump is fitted in the starboard engine room, for use in port to pump makeup feed water from the reserve feed tanks into the main feed tanks.

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*See Table II.

**Main Fire and Bilge Pumps.*—Located aft, on the outboard bulkhead, in each engine room, are two Blake, vertical, double-acting, single, fire and bilge pumps. They are arranged to draw water from the drainage system and the sea, and discharge to the fire main, sanitary system and overboard.

**Pipe Insulator Circulating Pumps.*—In each engine room there is a Blake, vertical, double-acting, single pump, for circulating sea water around the main steam-pipe flanges at bulkheads near magazines, to prevent the transmission of heat through the ship's structure to the magazines.

FORCED–LUBRICATION SYSTEM.

All working and moving parts of the main engines, except the valve links and valve-stem guides, are fitted with forced lubrication.

The installation in each engine room comprises three pumps,* two 500-gallon oil-settling and cooling tanks, together with the necessary piping and fittings. The settling and cooling tanks are provided with steam coils and cooling coils, the cooling agent being sea water, supplied from the pipe insulator circulating system. Each engine-room system is complete and independent, a cross connection being provided, however, for emergencies. The crank pits are of oiltight construction, fitted with an oil-drain well at the forward end, and so designed that no bilge water or dirt can enter therein. An oil trough is thus formed at the base of each engine to catch all the oil. A small pump* is provided in the starboard engine room, with suction connections to the bottom of each crankpit oil well, for pumping out any water that may have collected from the oil or through leakage. This pump discharges to the bilge. The entire engine is incased with a light galvanized sheet-steel casing, to prevent splashing and waste of oil, which is carried up to within about eighteen inches of the bottoms of the cylinders.

*See Table II.

The plant operates as follows: One pump draws the oil from the settling and cooling tanks, and discharges same through a pipe, fitted in duplicate for emergencies, having branches to the main bearings through holes in the caps. An annular groove in the center of each main bearing provides for the proper distribution of the oil, part of which lubricates the bearings, the balance passing through a radial hole in each journal, in wake of the groove, to the crankshaft axial holes, the openings at ends of each axial hole being closed by oiltight cover plates. From the axial holes the oil is forced through radial holes to the eccentric straps and crank-pin axial holes, and through radial holes to the surface of the crank pins. The crank-pin bearings are similarly grooved to the main bearings, all oil not used for crank-pin lubrication passing up through brass tubes secured to the connecting rods, to the crosshead bearings. After performing its function the oil escapes at the ends of the bearings and drains to the crank pit, where it collects in the drain well and is pumped by a second pump, through oil filters, to the main supply and settling tanks, thus completing its cycle, which is indefinitely repeated as described. The third pump is a stand-by for emergencies and connected to perform the duties of either of the other two pumps.

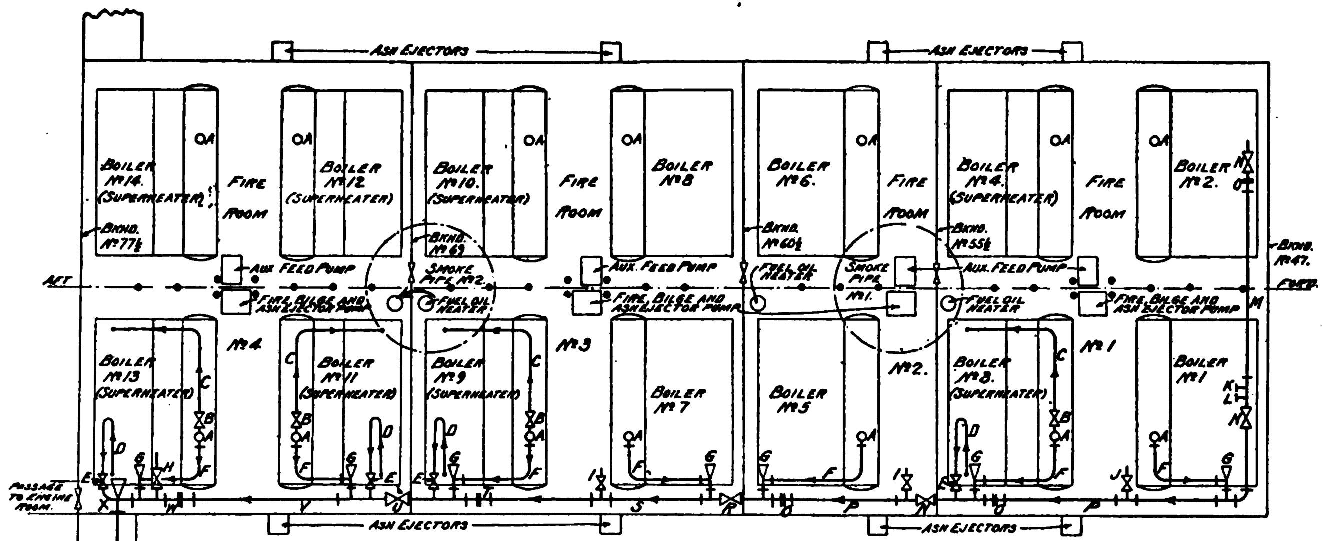
Pressure gages and thermometers are provided at each main bearing. The system is generally operated at about 50 pounds pressure.

Branches are taken off the system for the thrust bearings and main circulating-pump engine.

Large storage tanks of a total capacity of 2,000 gallons are installed in the engine rooms for making up leakage and other waste, and for replenishing the system when desired.

BOILERS.

There are fourteen Babcock & Wilcox water-tube boilers, arranged in four separate watertight compartments, as shown in Plate II. There are four boilers each in firerooms Nos.



ITEM	DESCRIPTION	ITEM	DESCRIPTION	ITEM	DESCRIPTION
A	5½" Boiler Stop Valve	I	4½" Aux. Steam Cross Connection	R	9" Globe Stop Valve
B	5½" G.S. Valves Steam to Superheater	J	3" Do.	S	10½" Main Steam
C	5½" Steam to Superheater	K	5½" Steam to Ford. Dynamos	T	10½" Expansion Joint
D	5½" Steam, Superheater to Main Line	L	4½" Steam to Windlass	U	12" Globe Stop Valve
E	5½" G.S. Valve, Do.	M	7" Main Steam Cross Connection	V	14" Main Steam
F	5½" Steam, Boiler to Main Line	N	7" Globe Stop Valve	W	14" Expansion Joint
G	5½" A.S. Valve, Do.	O	7" Expansion Joint	X	14½" Angle Stop Valve
H	7" Aux. Steam Cross Connection	P	7" Main Steam	Y	14½" Main Steam to Engine Room

NOTE.—Steam Piping on Port Side Similar to That Shown.

U. S. S. "NEW YORK."—ARR'G'T. OF MAIN STEAM PIPING AND MACHINERY IN BOILER ROOMS.—PLATE II.

1, 3 and 4, and two boilers in fireroom No. 2. Eight of the boilers, Nos. 3 and 4, and 9 to 14, inclusive, are fitted with superheaters.

The boilers are designed to operate the entire machinery plant at full power, with an average air pressure in the ash pits of not more than two inches of water. They are equipped primarily for burning coal, but are provided with an auxiliary installation for burning fuel oil in combination with coal.

The uptakes are of the usual design, and there are two smoke pipes, each 92.17 feet high above the grates. The forward smoke pipe is 11 feet 2 inches and the after one 12 feet 9 inches inside diameter, the former serving six and the latter eight boilers. Both smoke pipes are 13 feet 5⅜ inches outside diameter.

Boiler Data.

Number		14
Pressure, working, pounds per square inch		295
test, pounds per square inch		450
Height to top, external, feet and inches	Twelve of	12–00⅞
	Two of	11–09¾
Length on floor, feet and inches	Twelve of	10–01
	Two of	9–01½
Width on floor, feet and inches		17–02½
Drum, diameter, inside, inches		42
length, feet and inches		17–09⅞
thickness, inch		00$\frac{11}{16}$
Number of furnaces, each boiler		1
furnace doors, each boiler		4
Grates, length, feet and inches		7–00
width, feet and inches		15–10.32
per cent. of air space through		49
Total grate surface, square feet		1,554
heating surface, generating, square feet		62,213
superheater, square feet		3,267
Ratio, G.S. to H.S. (generating)		1 to 40.03
Number of tube headers, each boiler		27
2-inch tubes, each boiler		863
4-inch tubes, each boiler		29
Distance between headers, feet and inches	Twelve of	9–00
	Two of	8–00
Area through smoke pipe, forward, square feet		97.94
aft, square feet		127.68

G.S. ÷ area through smoke pipe, forward		6.687
	aft	6.955
Kind of forced draft		Closed firerooms.
Oil burners, number each boiler		6
	type	Peabody.
Diameter of	main steam stop valve, inches	05½
	main and auxiliary feed, stop and check valves (one of each), inches	02½
	surface-blow valve, inches	01½
	bottom-blow valves (two), inches	01½
	safety valve, twin, inches	04½

FUEL-OIL SYSTEM.

In addition to the usual coal-burning equipment, a complete oil-burning system is provided, for use as auxiliary fuel to be burned as required in combination with the coal.

The plant consists of two heavy-pressure, vertical, double-acting, single, pumps,* one in each engine room to the center-line bulkhead. These pumps draw the fuel oil from the double-bottom tanks, and deliver same to the oil burners on the boilers. Each boiler is fitted with six oil burners, mechanically atomized, of the Peabody type.

A 4½-inch suction pipe leads from the bottom of each tank to the suction manifolds located near the pumps in each engine room. The pump suctions from manifolds are 5 inches in diameter each, and are each fitted with a Macomb strainer. The discharge from each pump is 3½ inches in diameter, uniting into a common 3½-inch pipe (either pump being capable of supplying the entire system) which leads to the firerooms. The 3½-inch main is fitted with a Lalor automatic stop valve at junction with discharge pipes from both pumps.

In the firerooms the main is led forward, gradually decreasing in size from 3½ to 2 inches. There is a 1½-inch branch, taken off the main in each fireroom, leading to the fuel-oil heater in same fireroom. Each heater has 25.1 square feet of heating surface, the heating agent being live steam. By-passes are provided around the heaters. Each heater discharge pipe is 1½ inches, dividing into ¾-inch branches, fitted with

*See Table II.

Lalor automatic stop valves, to the burners on individual boilers.

Each fuel-oil tank is provided with a 1½-inch steam connection led to the bottom of the tank, for boiling out, and a 1-inch steam fire extinguishing connection at the top of tank.

Provision is made whereby the fuel-oil pump can transfer the fuel oil, through the filling pipe, to another vessel in emergency.

For washing out the tanks, one fire and bilge pump in each engine room is arranged for pumping their contents overboard, the connection for flooding and pumping out being portable and only connected up when cleaning the tanks, so there is no danger of flooding the tanks with sea water.

PNEUMERCATOR SYSTEM.

For ascertaining at any time the amount of fuel oil in the various double-bottom tanks, the Parks Pneumercator system is installed.

The installation consists of a small semi-spherical balance chamber placed at a predetermined location near the bottom of each tank. The interior of the balance chamber is in communication with the tank, through a hole in the side of the chamber, and a small pipe connects the top of the chamber with its recording instrument in the engine room at the fuel-oil manifold. The recording device consists of a glass mercurial tube provided with scale, calibrated to suit the tank to which it is connected.

The instrument operates on the following principle. The pressure due to the head of oil in the tank compresses the air in the chamber and connecting pipe line, which in turn causes the mercury to rise or fall in the tube in direct ratio to the pressure exerted. The tank contents is read off on the scale, as in the case of an ordinary thermometer.

In order to guard against overflowing the tanks when taking aboard fuel oil, an annunciator is installed in connection

with the recording device for indicating and signaling when any tank is 95 per cent. full.

The report of the engineer officer, on the operation of the instrument during the first six months of the commissioning of the vessel, pronounces it entirely satisfactory. A careful checking of its readings on various tests aboard the vessel have shown that it is thoroughly reliable. During the recent final-acceptance trials, when running the two-hours full-power run, burning oil and coal, the oil was measured by special measuring tanks on deck, which record agreed exactly with the pneumercator readings.

FIREROOM AUXILIARIES.

Forced-Draft Blowers.—Firerooms Nos. 1, 3 and 4 have four forced-draft blowers each, and fireroom No. 2 has but two. They are located in specially constructed blower rooms just below the protective deck and above the working flat in front of the boilers.

The fans are of the Sturtevant multivane type, and each is electric driven by a direct-connected 26-H.P. Diehl motor, controlled from the fireroom working level and the blower room at will. Air is supplied from the fireroom ventilators, which are closed at the bottom when under forced draft. The blowers are designed to run at 650 to 825 r.p.m.

**Auxiliary Feed Pumps.*—There are four Blake, vertical, double-acting, single, auxiliary feed pumps, one in each fireroom. They are arranged so that any pump can feed any boiler.

**Fire, Bilge and Ash-Ejector Pumps.*—In each fireroom there is a Blake, vertical, double-acting, duplex pump for fire, bilge and ash-ejector service. They are arranged to draw from the drainage system and sea, and discharge to the fire main, sanitary system, ash ejectors and overboard.

Ash Hoists.—The port ventilator in each fireroom is fitted for hoisting ashes, the necessary bucket guides, wire ropes,

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*See Table II.

sheaves, etc., being installed for the purpose. The hoists are operated from the main deck, at which deck the ash chutes are located at the ship's side.

The four ash-hoist engines, one for each hoist, are of the two-cylinder, reversible type, made by the Hyde Windlass Co. They are located in the upper fireroom hatch and are of the following principal dimensions:

Number of cylinders, each	2
Diameter of cylinders, inches	4½
Stroke, inches	4½

Ash Ejectors.—In addition to the ash hoists, there are eight 6-inch hydraulic ash expellers, two in each fireroom, port and starboard, discharging the ashes through scuppers above the protective deck.

MAIN STEAM PIPING.

The main steam piping is arranged in two symmetrical systems, one on each side of the vessel. The two lines are cross-connected in the forward fireroom and in the engine rooms by 7- and 9-inch connections, respectively. The branches from the boilers are 5½ inches in diameter each, and the lines proper are 7 inches in the forward fireroom, increasing to 9, 10½, 12 and 14 inches at each successive boiler connection. The pipes are increased to 14½ inches through the pipe passages between the engine and firerooms, reducing again to 14 inches in the engine rooms beyond the separators, which size continues to the throttle valves.

The arrangement of valves in the main steam piping is shown in Plates I and II.

AUXILIARY STEAM PIPING.

From the main steam pipe in the engine rooms is a 6-inch connection, with stop valve, which leads aft through the engine rooms and through the center-line bulkehad, with stop

valve on either side of same, forming a connecting loop between the two sides of the ship. From this pipe steam connections are taken for the various engine-room auxiliaries, steering engine and heating system aft. For use in port, when steam is shut off the main steam lines to engine rooms, there is a 4-inch auxiliary steam pipe, starboard side only, with stop valve at each end, from the auxiliary steam cross-connection in the after fireroom, which supplies steam to the auxiliary steam line in the engine rooms.

In the firerooms the auxiliary steam piping consists of a cross-connection, with stop valve at each end, between the port and starboard main steam lines, from which the branches to the auxiliaries are taken. The cross-connections are 3, 4½ and 4 inches in diameter, respectively, for firerooms Nos. 1, 2 and 3, and 7 inches in fireroom No. 4, which supplies the after dynamo room and distilling apparatus.

The forward dynamos and windlass take their steam off the main steam cross-connection in the forward fireroom.

Stop valves are fitted in all branches and sub-branches as required.

AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe is fitted throughout the machinery spaces and elsewhere as required for the various auxiliaries. Connections are provided to direct the exhaust steam into either main or auxiliary condensers, either feed-water heater, or into the atmosphere through the after escape pipe at will. There are also connections for admitting the exhaust steam to the L.P. receivers.

Stop valves are fitted in all branches at the main.

MAIN AND AUXILIARY FEED SYSTEMS.

Each main feed pump has a 5½-inch independent suction from the main feed tank in same engine room. The pumps discharge *via* the feed-water heaters, or by-pass same if desired, to the boilers. The combined discharge from each pair

of pumps is 6 inches, uniting in the after fireroom into an 8-inch connection leading forward and diminishing in size as it advances to 5 inches in the forward fireroom.

The auxiliary-feed suction main is taken off the 8-inch feed tank's cross-connecting pipe. This main is 9 inches in diameter, reducing in size to 5½ inches, as it leads forward to the auxiliary feed pumps, each pump having a 5½-inch suction connection. The auxiliary feed pumps discharge direct to the boilers in their respective compartments, or into the main feed line to any boiler. The discharge connections are 4½ inches in diameter at the pumps.

All branch, main and auxiliary feed pipes to the boilers are 2½ inches in diameter.

INTERIOR COMMUNICATION.

The customary engine and fireroom telegraphs, gongs, time-firing device, telephones, voice tubes, etc., are fitted for transmitting orders and signaling to the various machinery compartments and other parts of the vessel.

AIR-COMPRESSOR PLANT.

Located in each engine room are four 11-inch by 11-inch by 12-inch water-cooled Westinghouse steam-driven air compressors and two air reservoir tanks of about 45,000 cubic inches capacity each, for use in running pneumatic tools in the engineering department, blowing soot off the boiler tubes and for the gas-ejecting system for the guns.

Each compressor has a capacity of about 360 cubic feet of free air per minute at 150 pounds pressure.

A pneumatic main, independent of the gun gas-ejecting system, is led throughout the machinery space, with branches to the general workshop, evaporator and dynamo rooms, from which the connections for pneumatic tools and blowing soot off boiler tubes are taken.

EVAPORATING AND DISTILLING APPARATUS.

This plant is located on the protective deck, just forward of barbette No. 3. There are four evaporators, two distillers, two feed-water heaters, two distiller circulating pumps,* two evaporator feed pumps* and two fresh-water pumps,* together with their accessories. The plant has a combined capacity of 28,000 gallons of water per 24 hours, and is arranged to operate in double effect.

Evaporator Data (each).

Type	Reilly, horizontal, multi-coil.
Diameter, inside, feet and inches	5–01½
Length, over all, feet and inches	5–02½
Coils, number	36
diameter of pipe, inch	01
thickness of pipe, inch	00.065
Heating surface, square feet	221.4
Diameter of steam connection, inches	03
vapor nozzle, inches	04½
feed valve, inches	01¼
blow valve, inches	02½

Distiller Data (each).

Type	Reilly, vertical, multi-coil.
Diameter, inside, inches	31
Length, over all, feet and inches	4–03⅜
Coils, number	15
diameter, inch	01
thickness, inch	00.065
Cooling surface, square feet	92.25
Diameter of circulating water inlet and outlet, inches	04½
vapor inlet, inches	04½
drain, inches	02

Evaporator Feed-Water Heater Data (each).

Type	Bureau, made by Griscom-Russell Co.
Tubes, number	28
diameter, inside, inch	00¾
thickness, inch	00.065
Heating surface, square feet	19
Diameter of feed inlet and outlet, inches	02
vapor inlet and outlet, inches	04½

*See Table II.

GENERAL WORKSHOP.

A well-equipped machine shop is located amidship, on the protective deck, between the engine-room hatches.

The following machine tools, each driven by its own electric motor and all up to date and complete with the most modern attachments, are installed as listed below:

No.	Description.	Make and H.P. of Motor.
1	28-inch by 48-inch swing extension-gap lathe; E. Harrington and Sons Co.	Reliance, 7½ H.P.
1	14-inch swing lathe; The American Tool Works Co.	Reliance, 3 H.P.
1	14-inch swing lathe; The American Tool Works Co.	Reliance, 2 H.P.
1	16-inch column shaper; John Steptoe Shaper Co.	Reliance, 2 H.P.
1	31-inch simplex radial drill; Dreses Machine Tool Co.	Reliance, 2 H.P.
1	16-inch sensitive drill; Willey	Reliance, ½ H.P.
1	Universal milling machine; The Hendey Machine Co.	Reliance, 1 H.P.
1	Floor grinder; Willey	Reliance, 1 H.P.
1	Portable cylinder-boring machine; H. B. Underwood and Co.	2 H.P.
1	Planer	1½ H.P.
1	Bridgeport grinder	1 H.P.

BLACKSMITH SHOP.

A blacksmith shop is provided on the main deck, in the deck house, abreast of the after smoke pipe, port side. It is equipped with one portable and one permanent forge, together with anvil and all necessary tools and fittings. The permanent forge is fitted with an electrically-driven blast fan.

FOUNDRY.

A small foundry is installed in the deck house on the main deck. The foundry outfit consists of a small oil-burning crucible furnace of the Bureau's standard type, together with adequate allowance of crucibles and the necessary apparatus for handling and pouring the metal.

ELECTRIC PLANT.

There are two dynamo rooms, one just forward of the forward boiler room and the other just aft of the after boiler room. They are on the lower platform level, with their condensing apparatus below in a separate room in the hold, there being an access hatch and ladder between the two levels.

The distribution rooms, two in number, are located on the upper platform, one over each dynamo room. They contain the lighting, power and searchlight distribution boards only. The generator boards are in the dynamo rooms.

The generator installation consists of four 6-pole, compound-wound, 300-kilowatt, General Electric generators, two in each dynamo room, each driven by a two-stage horizontal Curtis turbine. Each generator will deliver at normal load 2,400 ampères of current at 125 volts, when running at 1,500 revolutions per minute. The generators are capable of delivering one-third overload for two hours without injury.

There is one condenser for each pair of generators of the same general design as the main and auxiliary condensers. Each condenser has its independent air pump,* centrifugal circulating pump,* hotwell-tank pump,* and a hotwell tank.

Dynamo Condenser Data.

Number	2
Inside diameter, feet and inches	4–05½
Thickness of shell (steel), inch	00⅜
Length between tube sheets, feet and inches	8–05⅜
Thickness of tube sheets, inches	01⅜

*See Table II.

Tubes, number	1,745
diameter, outside, inch	00⅝
thickness, inch	00.065
Cooling surface, square feet	2,400
Diameter of exhaust nozzles (2), inches	20
air-pump suctions, inches	07
circulating-water inlet and outlet, inches	08

Dynamo-Condenser Air Pumps Data.

Number	2
Type	Blake, vertical, twin, beam, single steam cylinder.
Diameter of suction nozzle, inches	07
discharge nozzle, inches	06
steam cylinder (1), inches	09
water cylinders (2), inches	18
Stroke, inches	12

Dynamo-Condenser Circulating Pumps and Engines Data.

Number	2
Type	Double-inlet centrifugal, driven by single engine.
Diameter of impeller, inches	26
suction (2), inches	05¾
discharge, inches	08
steam cylinder, inches	06
Stroke, inches	05

TRIALS.

The *New York* was subjected to the following trials in order to prove her machinery, ability to attain the designed speed and to ascertain her fuel and water consumption at the various speeds:

(*a*) A progressive trial over a measured-mile course in deep water for standardizing the screws, extending from maximum speed down to a speed of about ten knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable. (Designed speed 21 knots, with the air pressure in the firerooms not exceeding an average of 2 inches of water, and the steam pressure in the H.P. steam chest not exceeding 265 pounds above the atmosphere.)

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*See Table II.

TABLE II – PUMPS AND CONNECTIONS – U.S.S. NEW YORK.

No.	Pumps	Size (Ins.)	Type	Suction Pipes From – (Ins.)		Discharge Pipes To – (Ins.)		Location
2	Main Air	(2) 14 x (2) 35 x 21	Blake, Twin, Vertical, Single-Acting	12	Condenser	10	Feed Tank	1 in each Eng. Room
2	Main Circulating	42 Impeller; 10 x 20 x 10 Eng.	Horizontal, Centrifugal; Vert. Compound Eng.	(2) 17 15½	Sea Main Drain	23 4	Condenser Water Service	Do.
4	Main Feed	15½ x 9¼ x 24	Blake, Vert., Double-Acting, Single	5½ 5½	Feed Tanks Channel Way Cross Connection	5	Main Feed Line	2 in each Eng. Room
4	Aux. Feed	Do.	Do.	5½ 4 4 2½	Feed Suction Pipe Ship's Sides (a) Reserve Fd. Tanks Hose Connection	5 3½ 2½	Main Feed Line Reserve Fd. Tanks and Each Boiler in Compt. with Pump.	1 in each Boiler Room
4	Fire and Bilge	12 x 10 x 18	Do.	5 5 1½ 4½	Sea Drainage Hose Connection Fuel Oil Tanks (c)	5 5 5 2½	Fire Main Overboard Sanitary Main (b) Hose Connection	2 in each Eng. Room
4	Fire, Bilge and Ash Ejector	14 x 8 x 12	Blake, Vert., Double-Acting, Duplex	6 5 1½ 2½	Sea Drainage Hose Connection Do.	5 5 5 4½ 2½	Fire Main Overboard Sanitary Main Ash Ejectors Hose Connection	1 in each Boiler Room
4	Forced Lub. Service	7 x 8 x 8	Blake, Vert., Double-Acting, Single	3½ 3½	Oil Tanks Oil Well (d)	3 2½	Settling Tanks (d) Main Bearings	2 in each Eng. Room
2	Forced Lub. Return	Do.	Do.	4	Oil Well	3½	Settling Tanks	1 in each Eng. Room
1	Oil Well Drain	2 x 1½ x 2½	Blake, Vert., Duplex, D-Act'g.	1	Oil Well	½	Bilge	Starb'd. Eng. Room
2	Fuel Oil Supply	14 x 9 x 12	Blake, Vert., Double-Acting, Single	5 5	Storage Tanks Ship's Sides	4½ 4½ 4½	Burner Supply Storage Tanks Overboard	1 in each Eng. Room
2	Aux. Air	7½ x 14 x 12	Do.	6	Aux. Condenser	4	Main Feed Tanks	Do.
2	Aux. Circulating	28 Impeller; 5 x 4 Engine	Hor., Centrifugal Vert., Single Eng.	5	Sea	5	Aux. Condenser	Do.
2	Dynamo Cond. Air	9 x (2) 18 x 12	Blake, Twin, Vert., Single-Acting	7	Dynamo Cond.	6	Dynamo Room Hot Well Tank	1 in each Dyn. Cond. Room
2	Dynamo Cond. Circ.	26 Impeller; 6 x 5 Engine	Hor., Centrifugal Vert., Single Eng.	(2) 5½	Sea	6	Dynamo Cond.	Do.
2	Dyn. Rm. Hot Well	4½ x 5 x 6	Blake, Vert., Double-Acting, Single	2½	Dyn. Room Hot Well Tank	2	Aux. Feed Suction Main	Do.
2	Evaporator Feed	Do.	Do.	2½ 2½	Sea Dist. Circ. Disch.	2	Evaporators	Evap. Room
2	Distiller Circ.	8 x 12 x 12	Do.	6½	Sea	6 4½	Distillers Fire Main	Do.
2	Distiller Fresh Water	4½ x 5 x 6	Do.	2½	Distillers	2 1½ 1½	Fresh Water Tanks Aux. Fd. Suction Reserve Fd. Tanks	Do.
2	Pipe Insulator Circ.	7 x 8 x 12	Do.	4½	Sea	4	Pipe Insulators	1 in each Eng. Room
1	Reserve Feed	4½ x 5 x 6	Do.	3	Reserve Fd. Tanks	2½ 2	Main Feed Tanks Air Comp. Circ. Wat.	Starb'd. Eng. Rm.
2	Shaft Bilge	9 x 6	Plunger, Single-Acting	4½ 4½	Eng. Room Bilge Shaft Alley	4½	Overboard	1 in each Shaft Alley
3	Brine Circ.		Hor., Cent., Motor Driven	3	Brine Suction	3	Brine Discharge	Main Ice Mach. Rm.
2	CO_2 Cond. Circ.		Do.	3	Sea	2½	CO_2 Condenser	Do.
1	Brine Circ.		Do.	1½	Brine Suction	1½	Brine Discharge	Forw'd. Ice Mach. Rm.
1	CO_2 Cond. Circ.		Do.	1½	Sea	1½	CO_2 Condenser	Do.
2	Flushing		Do.	6	Sea	5	Crew's Ind. Sanitary System	Compt's. D26-P&S.
2	Fresh Water	(3) 4 x 6	Triple Plunger, Motor Driven	2½	Fresh Water Tanks	2½	Fresh Water System	Windlass Eng. Rm.

(a) No. 3 Fire Rm. only. (b) One pump in each Eng. Rm. (c) Portable Connection. (d) One pump in each Eng. Rm.

(*c*) An endurance and coal-and-water-consumption trial of twenty hours in the open sea in deep water at a speed of 19 knots as nearly as possible. All necessary auxiliaries usually required under service cruising conditions, the distilling plant excluded, to be in operation.

(*d*) An endurance and coal-and-water-consumption trial of twenty hours in the open sea at an average uniform speed of 12 knots as nearly as possible, with conditions similar to Trial (*c*).

(*e*) A trial of two hours' duration at the highest speed attainable, burning coal and oil fuel in combination.

The fuel consumption was carefully measured on Trials (*b*), (*c*), (*d*) and (*e*), and also the water consumption on trials (*b*), (*c*) and (*d*).

Standardization Trial (*a*).—This trial was run on the measured-mile course off Rockland, Maine, on October 20, 1914. The weather conditions were excellent.

After completing twenty successful runs, the trial was interrupted on the twenty-first run, due to a hot valve-stem crosshead guide on the after L.P. cylinder, starboard engine, which necessitated slowing the engines, and the trial was discontinued.

On the following day a fog set in, obscuring the ranges and making it impossible to continue further standardization runs without a probable long delay; and as the data obtained were, in the opinion of the Board exceptionally accurate and sufficient for the construction of a satisfactory curve, it was considered that an indefinite delay to obtain further runs for the high spot on the curve was not justified.

The data obtained on the twenty runs over the measured mile are given in Table III, from which the curves, Plate III, were plotted.

The official speed and revolution curve gave the following mean revolutions per minute of the main engines as necessary to attain the speeds desired for trials (*b*), (*c*) and (*d*):

4

TABLE III - STANDARDIZATION TRIAL DATA - OCT. 20, 1914 - U.S.S. NEW YORK

No. of Run	Time on Course: Mins.	Time on Course: Secs.	Speed in Knots	Revs. to go One Knot: Star. Engine	Revs. to go One Knot: Port Engine	Revs. to go One Knot: Mean	R.P.M.: Star. Engine	R.P.M.: Port Engine	R.P.M.: Mean	I.H.P.: Star. Engine	I.H.P.: Port Engine	I.H.P.: Total
1	5	34.0	10.78	331.0	329.6	330.3	59.47	59.20	59.34	1422	1277	2699
2	5	35.5	10.73	331.0	332.7	331.7	59.25	59.57	59.41	1435	1474	2909
3	5	40.9	10.56	337.6	339.9	338.8	59.41	59.80	59.61	1404	1465	2869
Mean of Group			10.70						59.44			2847
4	4	41.7	12.78	325.4	332.4	328.9	69.31	70.80	70.06	2258	2109	4367
5	4	54.2	12.24	341.0	341.3	341.2	69.54	69.60	69.57	2144	2173	4317
6	4	45.6	12.61	331.2	324.4	327.8	69.59	68.16	68.88	2162	2096	4258
Mean of Group			12.47						69.52			4315
7	3	55.8	15.27	340.8	332.4	336.6	86.72	84.56	85.64	4036	3895	7931
8	3	56.0	15.25	338.8	340.2	339.5	86.16	86.51	86.33	3926	4445	8371
9	3	49.8	15.67	332.4	330.1	331.3	86.82	86.22	86.52	3970	4340	8310
Mean of Group			15.36						86.21			8246
10	3	30.2	17.13	348.6	350.1	349.4	99.54	99.98	99.76	6247	6726	12973
11	3	23.8	17.66	339.5	333.8	336.7	99.94	98.27	99.11	6292	6140	12432
12	3	34.3	16.80	353.3	349.6	351.4	98.91	97.88	98.40	6123	6194	12317
Mean of Group			17.31						99.10			12539
13	3	5.8	19.38	337.6	331.3	334.5	109.08	107.03	108.06	8464	8037	16501
14	3	15.6	18.14	356.8	354.3	355.6	109.46	108.70	109.08	8593	8426	17019
15	3	4.4	19.52	332.9	334.9	333.9	108.30	108.95	108.63	8507	8829	17336
Mean of Group			18.93						108.71			16969
16	2	57.1	20.33	367.6	363.1	365.4	124.51	122.96	123.73	12834	12654	25488
17	2	45.2	21.79	340.0	341.7	340.9	123.47	124.11	123.79	12695	13804	26499
18	2	58.5	20.17	368.5	365.5	367.0	123.90	122.89	123.40	12357	12807	25164
Mean of Group			21.02						123.68			25913
19	2	40.8	22.39	350.6	344.5	347.6	130.82	128.55	129.68	15954	15831	31785
20	2	53.1	20.80	379.1	373.7	376.4	131.40	129.52	130.46	15965	15928	31893
Mean of Group			21.60						130.07			31839

TABLE IV — U.S.S. NEW YORK — TRIAL DATA.

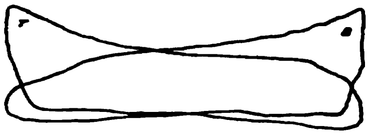
	4 HOURS' TRIAL (b)		20 HRS.' 19 KTS. TRIAL (c)		20 HRS.' 12 KTS. TRIAL (d)		2 HOURS' COAL & OIL BURNING TRIAL (e)	
[illegible] H.S.	[illegible] 0.747		[illegible] 0.575		[illegible] 0.576		[illegible] 0.4	
FUEL OIL, B.T.U. PER POUND	NOT USED		NOT USED		NOT USED		19,557	
SPECIFIC GRAVITY AT 60° F.	DO.		DO.		DO.		0.881	
POUNDS PER HOUR	DO.		DO.		DO.		17,509	
KNOT RUN	DO.		DO.		DO.		819	
POUND OF COAL	DO.		DO.		DO.		0.704	
MISCELLANEOUS DATA:								
SLIP OF PROPELLERS, PER CENT. OWN SPEED (MEAN)	15.25	15.25	11.67	11.67	8.79	8.84		
NUMBER OF BOILERS IN USE	14		14		8		14	
GRATE SURFACE USED, SQ. FT.	1,554		1,554		888		1,554	
HEATING SURFACE USED, SQ. FT.	62,213		62,213		36,116		62,213	
I.H.P. MAIN ENGINES PER SQ. FT. OF GRATE	19.104		11.727		4.777			
SQ. FT. OF H.S. PER I.H.P. MAIN ENGINES	2.096		3.414		8.514			
C.S. (MAIN COND.) PER I.H.P. MAIN ENGS.	0.8154		1.433		6.178			
KNOTS PER TON OF COAL	1.035		1.35		2.375			
CUT-OFF:								
H.P. CYLINDER	FULL	FULL	0.68	0.68	0.56	0.56		
I.P. DO.	DO.	DO.	0.70	0.70	0.63	0.63		
F.L.P. DO.	DO.	DO.	FULL	FULL	0.57	0.57		
A.L.P. DO.	DO.	DO.	DO.	DO.	0.57	0.57		

* NOT RELIABLE.

INDICATOR CARDS SET No. 7 - U.S.S. NEW YORK - FOUR HOURS' TRIAL.

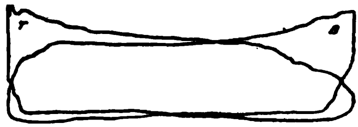
STARBD. ENGINE (129.22 R.P.M.)

H.P. CYL.
M.E.P. 108
M.R.P. 12.15
I.H.P. 3951
SCALE OF SPRING 150

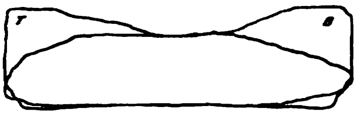


PORT ENGINE (129.21 R.P.M.)

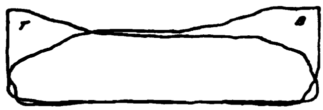
H.P. CYL.
M.E.P. 122
M.R.P. 13.23
I.H.P. 4463
SCALE OF SPRING 150



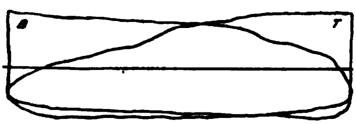
I.P. CYL.
M.E.P. 47.7
M.R.P. 14.06
I.H.P. 4618
SCALE OF SPRING 60



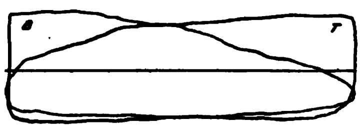
I.P. CYL.
M.E.P. 47.5
M.R.P. 13.63
I.H.P. 4598
SCALE OF SPRING 60



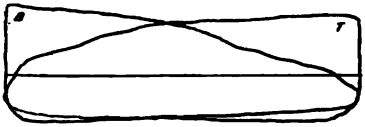
F.L. P. CYL.
M.E.P. 17.8
M.R.P. 8.9
I.H.P. 3002
SCALE OF SPRING 20



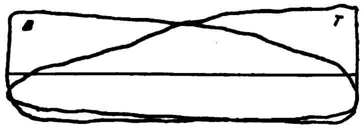
F.L.P. CYL.
M.E.P. 18.5
M.R.P. 9.3
I.H.P. 3120
SCALE OF SPRING 20



A.L. P. CYL.
M.E.P. 18.6
M.R.P. 9.3
I.H.P. 3111
SCALE OF SPRING 20



A.L.P. CYL.
M.E.P. 19
M.R.P. 9.5
I.H.P. 3204
SCALE OF SPRING 20



TOTAL M.R.P. 44.41
TOTAL I.H.P. 14682

TOTAL M.R.P. 45.66
TOTAL I.H.P. 15385

TOTAL I.H.P. BOTH ENGINES 30067

PLATE IV.

Speed, in knots.	R.p.m.
21	123.5
19	109.1
12	66.8

Twenty-Hour 12-Knot Endurance and Coal-and-Water-Consumption Trial (d).—This trial followed the standardization trial. It was commenced just before noon, October 21,

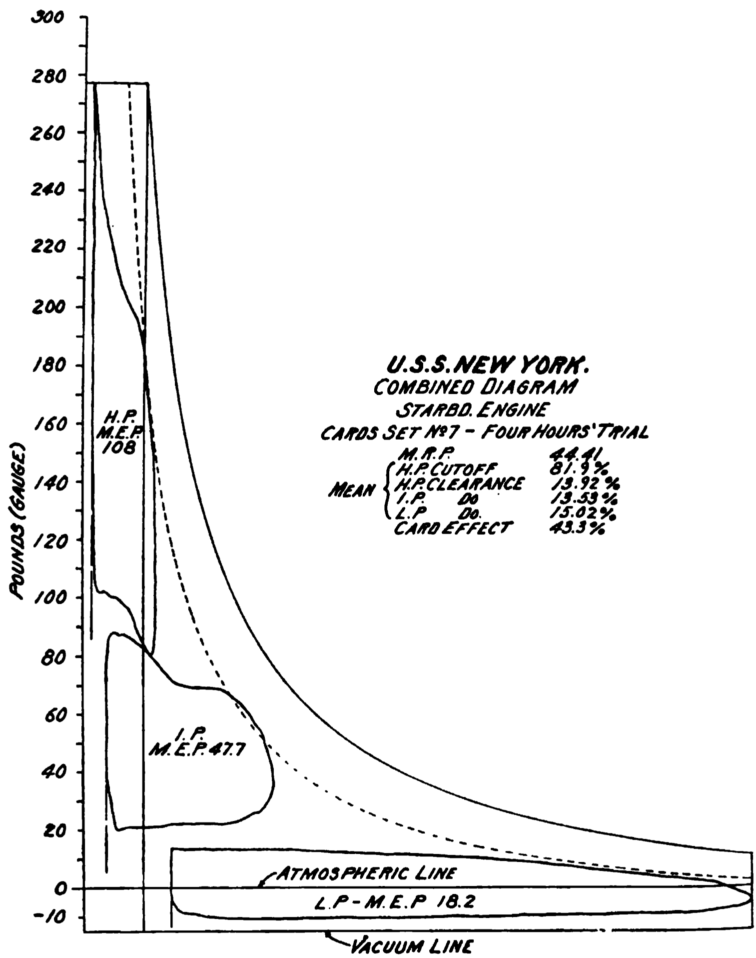


PLATE V

and satisfactorily concluded on the following day. The weather was fair with overcast sky and foggy throughout most of the trial. The average speed was 12.11 knots, and the fuel and water consumption and other data obtained are given in Table IV.

Four-Hour Full-Speed Trial (*b*).—The four-hour full-speed trial was begun at 11:30 A. M., October 22, off Nantucket Shoals Lightship, and was successfully concluded at 3:30 P. M. The weather was very good, with gentle breezes and smooth sea. The designed speed was easily exceeded, the average for the four hours being 21.47 knots.

Table IV gives the data obtained and Plate IV shows indicator cards, set No. 7, taken on this trial. The starboard cards of set are also shown combined in Plate V.

Twenty-Hour 19-*Knot Endurance and Coal-and-Water-Consumption Trial* (*c*).—Upon conclusion of the 4-hour full-speed trial, the 20-hour 19-knot run was immediately commenced. The trial began at 3:30 P. M., October 22, and was completed at 11:30 A. M., October 23. The trial was run in varying conditions of wind and sea, but not unfavorable. An average speed of 19.23 knots was maintained, and the data is given in Table IV.

Two-Hour Coal-and-Oil-Burning Trial (*e*)—This trial concluded the tests of the vessel, and was run immediately after the 20-hour 19-knot trial, on October 23, from 11:30 A. M. to 1:30 P. M. The weather was fair, with moderate sea and breezes. For data see Table IV.

All trials were conducted with the regular navy crew, the vessel being in commission when the trials were run.

TURBINE ELECTRIC PROPULSION OF A BATTLESHIP COMPARED WITH OTHER MEANS.

BY P. W. FOOTE, LIEUTENANT COMMANDER, U. S. N.

The introduction of forced oil lubrication for reciprocating engines so greatly removed the trouble of hot bearings at high speed that the turbine engine lost a great deal of its superior value for uses on a battleship, as in many other important points the turbine was no better and in a number not so good as the reciprocating engine. This was very ably demonstrated in a paper prepared by Captain C. W. Dyson, U. S. N., entitled, "Engineering Progress in the U. S. Navy," and read before the Society of Naval Architects and Marine Engineers, November, 1913 (see Volume 20 of the Transactions of the Society). In this paper certain requirements were tabulated which necessarily governed the selection of the propelling machinery of heavy vessels of moderate speeds, in which class a battleship falls. The U. S. S. *Delaware* was chosen as the model of a battleship with reciprocating engines, and her performances were compared with those of her sister ship, the *North Dakota*, driven by Curtis turbines, direct-connected to propeller shafts, and also with those of the U. S. S. *Utah*, having direct-connected Parsons turbines.

Captain Dyson's paper showed the reciprocating engines to be more advantageous for use in battleships in the U. S. Navy than the turbine, in the following characteristics:

(*a*) Greater economy in low speeds, hence greater cruising radius.

(*b*) Ease of up-keep of machinery, allowing greater readiness for duty.

(*c*) Efficient propellers for maneuvering, and high backing power.

(*d*) Weight and space.

(*e*) Minimum vibration of hull due to machinery, and steadiness of hull as a gun platform.

In the following particulars the reciprocating engine and the turbine were shown to be about equal:

(*a*) Economy at maximum speed when this does not exceed twenty-two knots.

(*b*) Reliability when driven at high powers.

(*c*) Boiler weight and space.

The only condition in which the turbine was considered superior to the reciprocating engine was in case the necessary power to be developed were greatly increased from that at present and if the ordinary cruising speed were made considerably higher than now used.

The above characteristics cover the important and controlling features of the propelling machinery for a battleship, and the discussion under the different headings showed good reason why reciprocating engines were preferred to turbines for the *New York*, *Texas* and the *Oklahoma.*

It being generally accepted that rotary motion for an engine is preferred to reciprocating, the question naturally arises, "Is there any way in which the turbine engine may be used for battleship propulsion and at the same time equal or improve the efficiency of the reciprocating engine in the above-mentioned particulars in which it is now considered inferior?" This question is a particularly important one, due to the increase in power now being required, causing the turbine to be superior in weight and space, as mentioned by Captain Dyson on page 58 of his paper.

The answer to the above question seems to depend upon finding an efficient means of transmitting the power of the steam turbine to the propeller shafts with little loss of power but with a great reduction of speed in r.p.m., this being due to the fact that it is fundamental in the "nature of the beast" that a steam turbine only reaches its greatest efficiency at a

very high speed in r.p.m., these speeds varying from 1,200 to to 3,600 r.p.m., depending upon the power developed, and, by the same token, the efficient speed of a propeller is very low, being about 100 to 125 r.p.m.

There are now two means, for which great merit is claimed, by which the speed of the turbine may be reduced and the power transmitted to the propeller with a high efficiency. These two means are (1) mechanical reduction gear, (2) electrical transmission, using electric generators and motors.

There is also the Föttinger hydraulic gear, proposed for this purpose, but it is probably only good for limited speed reduction. We will, therefore, only consider the other two.

The mechanical reduction gear is in use in the U. S. S. *Neptune*, where about 2,500 S.H.P. is transmitted on each of two gears from a Westinghouse-Parsons turbine running at 1,250 r.p.m. to the propeller running at 135 r.p.m. for maximum speed of 14 knots. This gear is of the modified Melville-Macalpine type, built by the Westinghouse Machine Company.

Most people are familiar with the design of this gear, and it is not necessary to give a description of it here.

The *Neptune's* reduction gear has proven satisfactory, and it has been recommended for further use in ship propulsion. There were some difficulties encountered with the *Neptune's* machinery, but the trouble was not with the reduction gear. This method of speed reduction has found great favor with Sir Charles Parsons, of England, and it was reported in Autumn, 1912, that there were upwards of 100,000 horsepower of Parsons geared turbine machinery built and under construction.

Mr. Charles Curtis also favors the use of the mechanical reduction gear for ship propulsion, and in his discussion of Captain Dyson's paper he stated that in his judgment the best form of propelling machinery for the next United States battleship would be straight turbines, combined with small cruising geared turbines, the latter to be used only at low power, and on long cruises where economy of coal was of prime importance. When the cruising turbines were in use,

the steam would pass first through the geared turbine and then through the main turbines. He predicted a large gain in economy, "at least 25 per cent.," at speeds of 10 or 12 knots. It is interesting to note that the design of the *Pennsylvania* follows almost exactly the above outline, and her performances will be awaited with much interest.

On March 13, 1913, Sir Charles Parsons read a paper on "Mechanical Gearing for the Propulsion of Ships" before the Institute of Naval Architects (London), the effect of the paper being to give an account of the progress made in developing mechanical gear.

From this paper the following information is obtained: In England there are in actual service cargo steamers, channel steamers and warships using a total of 26,000 H.P. developed by steam turbines and transmitted by the reduction gear. There is under construction 120,000 H.P. of turbine machinery with mechanical reduction gear, of which two installations will be over 20,000 H.P. each. The number of turbine reduction-gear units to transmit this 20,000 H.P. is not stated. It is probably divided into four units, judging by other data given in Sir Charles' paper.

The S. S. *Normannia* and the S. S. *Hantonia*, of 1,900 tons displacement, with a S.H.P. of 5,000 at service speed of 18 knots, operating in the channel service for the London and South-Western Railway Company, have reduction gearing, and they show an economy 40 per cent. greater than the other turbine steamers on the same service, this partly due to increased efficiency of turbines, partly to increased efficiency of propellers due to the lower speed, and partly due to the improved form of vessel incidental to the reduction of boilers and adoption of twin screws—all being due to the use of the reduction gear. After 26,000 knots the *Normannia's* gearing was found to be in perfect condition.

A cargo steamer, the *Cairnross*, sister ship to the S. S. *Cairngowan*, having reciprocating engines, has been fitted with turbine reduction gear developing 1,600 S.H.P., and a coal-consumption trial, in which the coal was same quality and

measured the same way on both ships, the two ships running side by side showed the turbine-reduction gear ship to be 15 per cent. more economical than the one with reciprocating engines.

Sir Charles states that so far no limit in regard to the surface speed of the teeth has been discerned, and that there is no evidence of any limit of power that can be transmitted by mechanical gearing. This applies to the design of the gear itself. There are certain conditions of weight and ease of control which will be mentioned later, which do limit the *application* of mechanical gearing.

The chief difficulty to be overcome in mechanical gearing is to protect the teeth of the gear from unequal and excessive strain due to lack of absolutely perfect alignment of the shafts and their necessary "play" in their bearings. This was accomplished in the Melville-Macalpine gear by the "floating bearings," which are a complicated piece of mechanism. Mr. Parsons states that a means of cutting of double helical teeth has been obtained which, when used with flexible couplings between turbine shafts and their pinions, renders unnecessary the floating bearings.

This last statement is substantiated by experiments conducted by the General Electric Company, who have developed a mechanical gearing which has given extremely satisfactory results without the floating bearings. The degree of the development of this gearing may be illustrated by stating that they recently proposed a turbine and reduction gear for the torpedo-boat tender *Melville* to develop and transmit 4,000 H.P., turbine speed 2,400, propeller speed 110. The weight of the machinery was guaranteed not to exceed 55 tons. This furnished an interesting comparison with the *Cyclops*, *Neptune* and *Jupiter*, as follows:

Cyclops.—Reciprocating engines; weight of machinery, 280 tons.

Neptune.—Turbine with Westinghouse reduction gear; weight of machinery not published.

Jupiter.—Turbine electric; net machinery, 156 tons. The above ships have about 5,500 I.H.P. with twin screws.

The above proposal for the *Melville* was for 4,000 H.P. and single screw, the difference in weight being due to one screw instead of two—1,500 less H.P.—lighter reduction gear due to absence of the floating bearings and lighter turbine per H.P., due to its higher speeds.

This comparison emphasizes the fact that for ships of this size and horsepower the turbine mechanical gear is more advantageous than the turbine electric gear, if the facility of control is not a paramount requirement, though both of these are superior to the reciprocating-engine machinery.

That the advantage lies with the turbine electric gear for ships of greater power and displacement will be shown later, this being particularly the case when facility of control is of great importance, as is the case in a battleship.

When the plans for the propelling machinery of the U. S. S. *Pennsylvania* were being considered, a design for turbine electric propulsion for this ship was made by Mr. W. L. R. Emmett, of the General Electric Company, and the company showed their faith in its successful operation by being willing to guarantee the characteristics, water rates, etc., described later, and in case of failure to meet the guarantees, the company would remove the turbine electric installation without cost to the Government. As the price of this installation would have been approximately $350,000.00, the faith of the company is evident.

For use in illustration and comparison, the following curves and sketches are herewith attached, numbered as shown:

No. 1. Plan of engine room, 31,000 S.H.P., with turbine electric installation as proposed by the General Electric Co.

No. 2. Plan of engine room of a large battleship, 60,000 S.H.P., as proposed by Sir Charles Parsons in his paper above mentioned.

No. 3. Comparison of water rates required to drive a 31,000-ton battleship with direct-connected turbine machinery and with turbine electric machinery, the water rates being referred

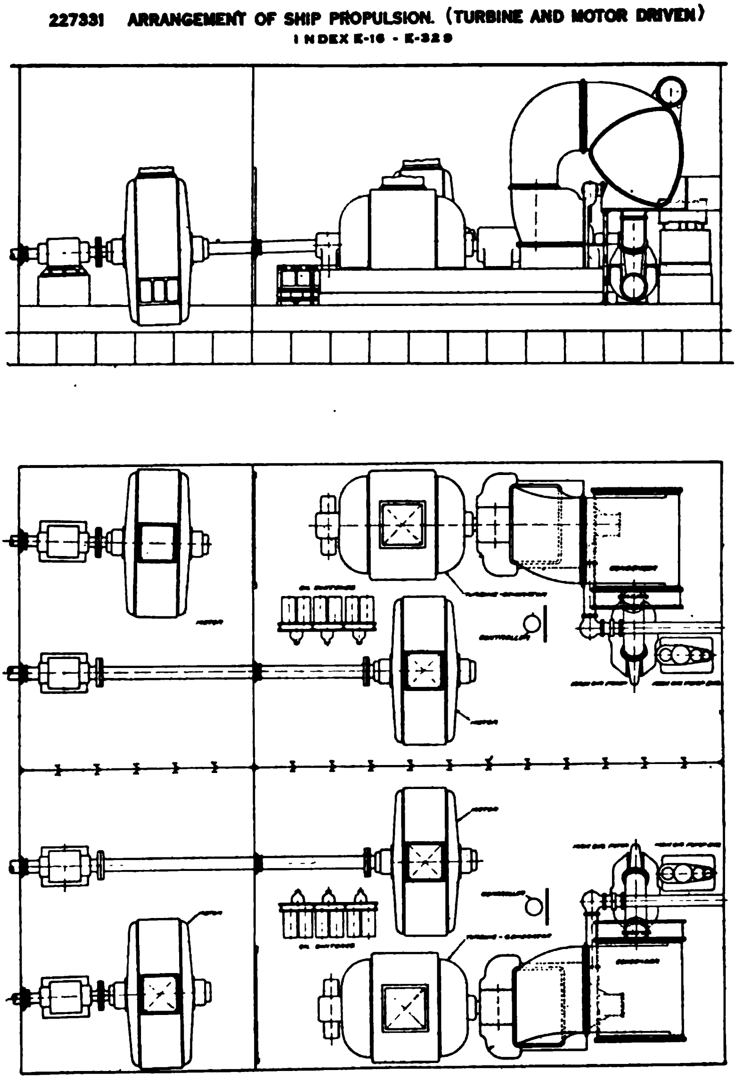
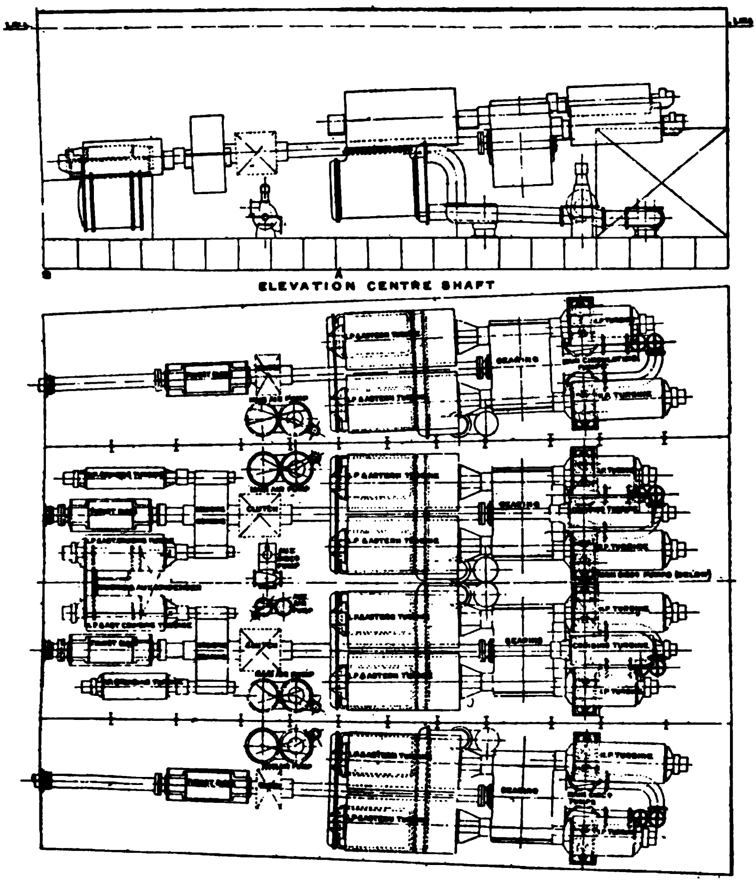


FIG. I.

to the power, 31,700, required for the propellers of higher speed. The curves show the relation between the amount of steam required per knot at the different speeds, these calculations being made by Mr. W. L. R. Emmett.

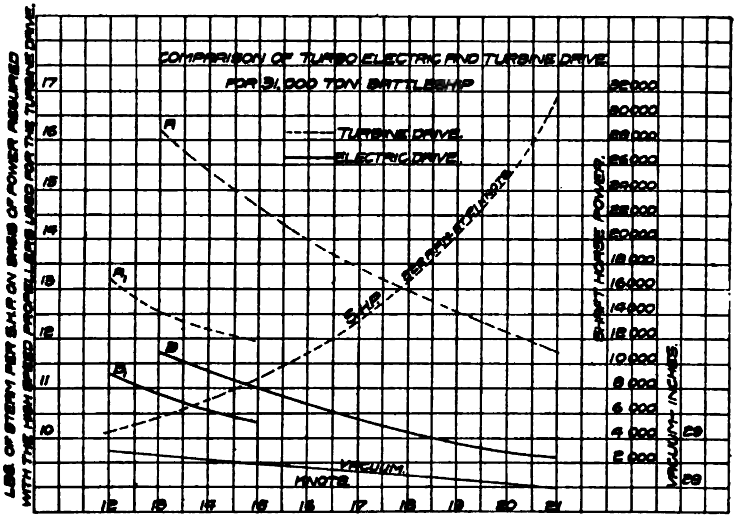


227332 . GEARED TURBINE MACHINERY FOR BATTLESHIP WITH SEPARATE CRUISING INSTALLATION.
INDEX E-16 - E-329

FIG. 2.

No. 4. Shows the relation of powers required to drive the ship at 21 knots with different propeller speeds and also the diameter of propeller for different r.p.m. at 21 knots.

No. 5. Is a curve published in London "Engineering" of December 29, 1911, in connection with a series of articles on the subject of "The Steam Turbine." This curve is the



A. — ESTIMATED RESULT WITH MAIN TURBINES ALONE AS ADOPTED.
A1 — " " " GEARED CRUISING TURBINES IN ACTION.
B — RESULTS ATTAINABLE WITH TURBO ELECTRIC DRIVE WITH TWO GENERATORS IN USE AND HIGH SPEED CONNECTION OF MOTORS EXPRESSED IN TERMS OF THE POWER REQUIRED FOR THE PROPELLERS ADOPTED FOR DIRECT TURBINE DRIVE.
B1 — RESULTS AT CRUISING SPEEDS WITH ONE GENERATING UNIT AND LOW SPEED CONNECTION OF MOTORS.

FIG. 3.

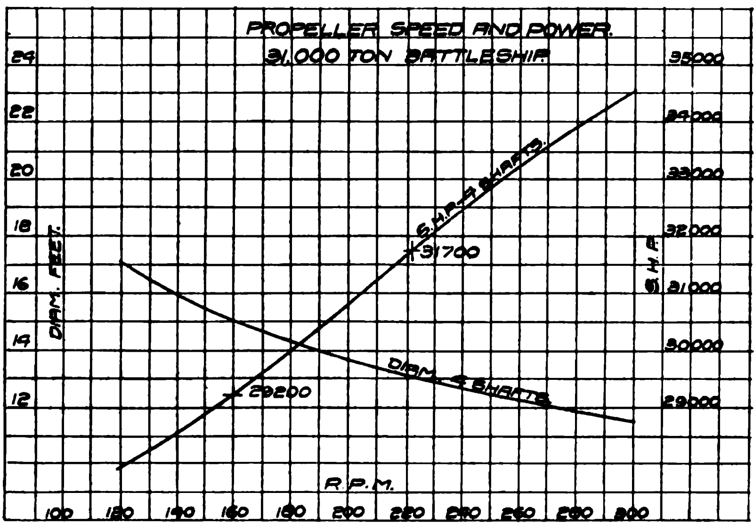


FIG. 4.

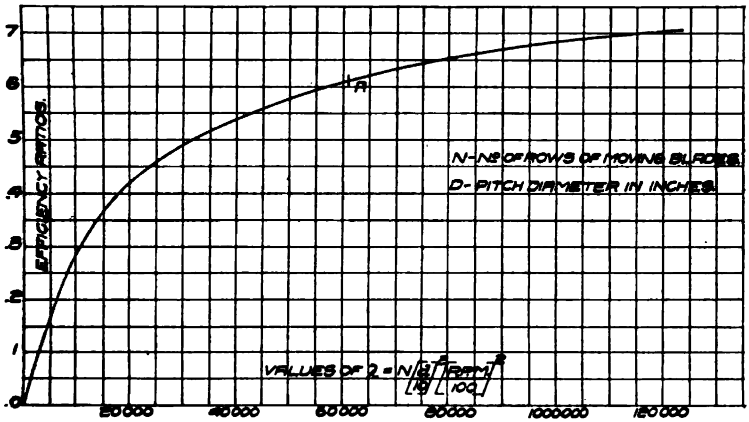


FIG. 5.

authority for the calculation as to the performance of the steam-turbine design. This curve shows limits of efficiency as governed by the number of rows of moving blades, the pitch diameter and the speeds.

The author of this curve states that he has found this curve to agree closely with performances of many turbines of different types, and that most of the points have been deduced from the trials of a set of very large and very efficient marine turbines, and that many high-speed turbines have been found to agree closely. This curve has, for some time, been used for approximate comparisons by different turbine designers, and in that way has been compared with the test performances of a large number of the best turbines of recent design, and in all of these only one or two came slightly above the curve.

The maximum possible efficiency of direct-connected turbines with cruising turbines having reduction gear with certain assumed numbers of blades and diameters should be indicated by the point marked "A" in the curve, and this value has been used in working Curve "A" for high speeds on blueprint No. 3. The curve marked "A_1", for cruising conditions, is obtained in a different way, but it is believed it shows even better rates than can be obtained with a design in which the geared turbines perform only half the work required, the

remainder being done under disadvantageous conditions in the main turbines.

The performance of the turbine for electric equipment shown on Plate 3 by Curves B and B–1, are taken from actual tests of a generating unit almost exactly similar to those proposed for the electric drive. The efficiency of the turbine proposed for the electric drive calculated by the formula on blueprint No. 5 is 75 per cent., which is beyond the limits of curve drawn.

Plate No. 6 shows a view in a section of a turbo-generator very similar to the two that would be used for electric drive. Due to the greater speed reduction obtainable with the electric drive, it will be noted from blueprint No. 4, that 160 r.p.m. would be used instead of 222, the diameter of propeller would be 15 feet instead of 13 feet, S.H.P. 29,200 instead of 31,700, due to more efficient propellers. Using the data from the above-mentioned curves, the following table shows the weights of the two equipments for a 31,000-ton battleship.

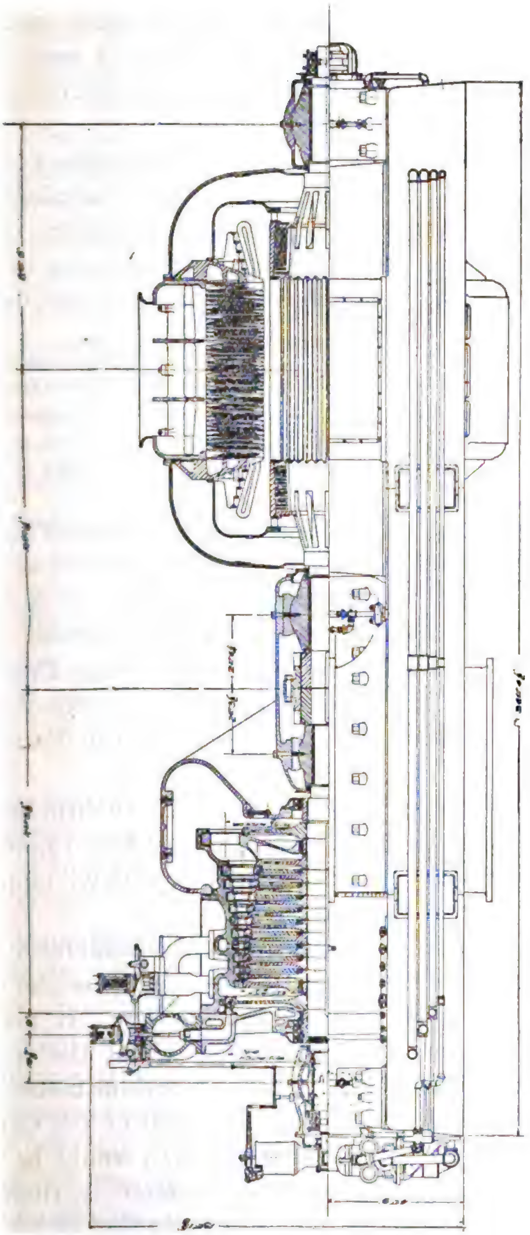
......	R.P.M. 21 knots.	S.H.P. required, 21 knots.	Pounds steam per hour turbines alone, 21 knots.	Pounds steam per hour turbines alone, 15 knots.	Weight driving machinery, in tons.
Turbine drive with geared turbine for low speeds	222	31,700	374,000	106,000	749
Turbine-electric drive	160	29,200	305,000	91,000	598

The individual weights being as follows for electric machinery:

One motor, 164,000 pounds; outside diameter, 172 inches.

One turbo-generator, complete, 320,000 pounds; the generator being 200,000 pounds and the turbine being 120,000 pounds.

With reasonable allowances for steam required outside of



224985 ASSEMBLY CROSS SECTION OF 10000 KW. 1500 R.P.M. HORIZONTAL STEAM TURBINE
INDEX E-16

FIG. 6.

main turbines, it would appear that the ship could, with the turbine-electric equipment, operate equally well with two boilers less than with direct turbine drive. If two boilers were omitted the whole weight saving would amount to 266 tons.

For comparison with the above, the following data is quoted from Captain Dyson's paper. The weights tabulated are given as "Engine-Room Weights," and probably include engine-room auxiliaries for which an allowance must be made in comparing with above figures which, are for driving machinery:

	Engine-room weights, dry tons.
Delaware	728.26
North Dakota	731.23
Utah	864.69

From the above comparison it appears that a great increase in H.P. may be obtained by using electric drive with a *decrease* in weight from that of the reciprocating engine, a characteristic in which this engine is superior to the turbine direct connected. As the increase from S.H.P. 21,528 for *Delaware* to S.H.P. 31,700, the calculated S.H.P. for a 31,000-ton battleship with direct-connected turbines, is obtained with increase in weight.

The machinery comprising the proposed turbine-electric drive is shown on Plate No. 1. It consists of two 12,000-kw. (approximate) turbo-generators, with a 7,500-kw. (approximate) motor on each of the four shafts.

It will be noted that this design differs considerably from that proposed heretofore for a battleship, and it is also quite different from the installation on the *Jupiter*. It may be noted here, in passing, that the *Jupiter* is not the type of ship that is best suited for electric drive, a turbine mechanical gear of type proposed by Mr. Parsons, or that of the General Electric Company proposed for the *Melville*, would be more desirable. The electric drive on the *Jupiter* is, therefore, more in the nature of an experimental installation rather than

being one that best fills the requirements for the propelling machinery of a ship of that type.

In the design formerly proposed for a battleship two motors were to be used on each shaft, this being necessary to meet conditions of going "ahead" at various speeds and at stopping and "backing," as to handle the load under these two conditions two motors of different characteristics were necessary, one of low reversing torque and a high running efficiency, and the other of high reversing torque by using external resistance to start under full load, the "running" motor was of the squirrel-cage type, the other was a polar-wound motor with slip rings and external resistance, this latter being used to absorb the high current when starting under load when a high torque is required, which is the condition when the ship is being maneuvered, backed, etc.

The motor now proposed is a double squirrel-cage motor, being an adaptation of the Boucherot type, and it is one having a most satisfactory combination of the characteristics of the two motors above mentioned. While this type of motor was designed by Boucherot in 1894 and 1895 with successful results, it has only been during the last six months that a motor of this type has been developed by the General Electric Company. A motor of this type has now been designed and tested by this company, and the test results agree so closely with the calculations that the company is now ready to build motors of this kind with the usual guarantees covering satisfactory service.

As the name implies, the double squirrel-cage motor (well described in "Electric Motors" by Henry M. Hobart, pages 325–336), has for its secondary two sets (outer and inner) of bars concentric with the core of the rotor, the primary being an ordinary stator of an induction motor.

The outer set of bars are of material having a high resistance and low inductance, and the inner set having a low resistance and high inductance. It is in this way that the characteristics of the two motors are combined into one.

The action is as follows: At starting, the current induced

into the high resistance (outer) bars exerts a high starting torque, a necessary requirement for backing the ship. As the motor comes up to synchronous speed the inner bars of low resistance carry the larger part of the current; the losses are small and the efficiency high. The application of this method, even in a small degree, makes it possible for the generator and the motor to pull into synchronism in a very brief period, after which full torque of the turbine for reversal is available, or, if desirable, provision can be made for ample torque without approaching synchronism.

This type of motor exactly fulfills the requirements for ship propulsion where such wide variation of loads, speeds and operating conditions are found. This motor also is adaptable to pole changing in the desired ratio, which is accomplished by a simple group of pole-changing switches in the stator circuits.

Print No. 7 shows the torque curve of this type of motor

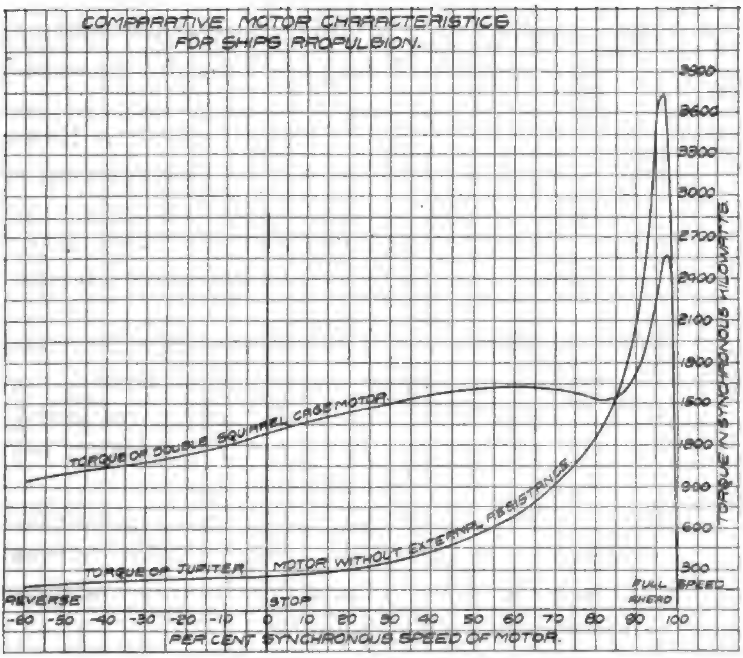


FIG. 7.

and also the torque curve of the *Jupiter's* motor when the external resistance is short-circuited, which is the "running" condition of the latter; with the resistance in circuit, a high "reversing" torque is obtained. The features of design, as mentioned above, of the double squirrel-cage motor allow a high torque under all conditions without the necessity of external resistance.

It is due to the fact that relative speeds of the motor and generator depend upon the relation between their poles that the electric units allow such a large reduction of speed between the turbine and propeller, thus permitting the turbine to run at its efficient speed, which is necessarily high, and the propeller to run at its efficient speed, which is necessarily low.

The speed of the motor is to the speed of the generator in an inverse ratio of their respective poles. On the *Jupiter*, for instance, the generator has two poles and the motors have 36, the speed reduction is 18 to 1. It is due largely to this fact that electric propulsion does not lend itself so readily to ships of small horsepower, as the size of the motors mechanically limits the number of poles, and therefore the speed reduction is more limited than in ships of large horsepower where large motors are used.

It seems that about 5,000 or 6,000 S.H.P. is about the low limit for electric propulsion; below these powers mechanical reduction gearing is generally more efficient and desirable unless facility of control is paramount. Each case must, therefore, be decided in accordance with requirements.

The poles of an induction motor depend upon its windings, and the windings can be so connected to a pole-changing switch or controller that the number of poles may be changed in a certain definite ratio as desired. This permits the ratio between the poles of the generator and of the motors to be varied to produce the most economical operation at the high and at the low speeds of the ship, and it is in this way that the economical rates as shown on Curve B and blueprint No. 3 are obtained.

For the electric drive shown on Plate No. 1, the turbine and generator would have a speed of about 1,900 r.p.m.; the generator would have been a bi-polar and the motor would have had 24 poles for speeds above 15 knots, and 36 poles below 15 knots, allowing speed reduction of 12 to 1 for high speed of ship and 18 to 1 for cruising speed. Minor variation in speed is obtained by varying speed of turbine by governor control.

The electric drive permits great facility of control either from the bridge or any other point, such as conning tower or any fire-control station, and in this point easily excels the reciprocating engine, the turbine direct-connected, or the turbine mechanical gear. The propeller may be started ahead, stopped, or run astern by turning the handles of a controller, which can also be used to control the turbine speed as desired. The turbine governor automatically regulates the speed of turbine when load is varied on the motor.

An advantageous point in this equipment is that, as the speed of the motor is synchronous with the generator, the propellers cannot race when the ship pitches them out of the water, the speed of the generator being controlled by a particular design of governor on the turbine. This not only saves the wear on the machinery, but saves the loss caused by the inefficient speed of the propeller when racing, which it is believed would make quite a difference in the fuel per knot in heavy weather.

By using the above-mentioned motor a starting torque of 100 per cent. full-load running torque may be obtained. This allows as high backing power as the design of propeller will permit, and in this way equals the reciprocating engine and far excels the turbine mechanical-reduction gear. The lack of sufficient backing power has been a very unsatisfactory point in our turbine ships. From Captain Dyson's paper the following is tabulated:

	Backing power in per cent. of ahead power.
U S. S. *Delaware*	87.5 per cent.
U S. S. *Utah*	35.7 per cent.

From trials, the *Delaware* steaming at 21 knots, and the *Utah* at 20 knots, the time taken to bring the vessels dead in the water were, for the *Delaware*, 1 minute, 52 seconds; the *Utah*, 4 minutes, 44 seconds. For reasons above mentioned the "electric" battleship would be equal to or superior to the reciprocating-engine ship in this regard.

From data tabulated in the JOURNAL OF THE SOCIETY OF NAVAL ENGINEERS, February, 1913, pages 83 and 84, it is seen that the *Delaware* at a speed of 21.56 knots per hour required as follows:

Total water, all purposes, per hour, pounds	422,931
Pounds of coal per knot per hour	2,502
Total of coal, 24 hours, tons	578

This data agrees closely with the data obtained from contractor's trials, when the pounds of coal per knot per hour were 2,640.

Using the above data it is found that the *Delaware* had a boiler evaporating factor of 9.5 pounds of water per pound of coal.

Using the factor of 9.5 pounds water per pound of coal, and taking the guaranteed water rate of 9.6 pounds water per S.H.P. for the electric battleship, and assuming 60,000 pounds water per hour for auxiliaries, which, according to data obtained from the *Florida* and *Utah*, is a fair figure, we find as follows, for speed of 21 knots:

Pounds water, main engines	= S.H.P. × water rate,
	= 31,700 × 9.6 per hour,
	= 304,000 pounds.
Pounds water for auxiliaries	= 60,000 pounds per hour.
Total water all purposes	364,000 pounds per hour.
Pounds of coal per hour (using 1 pound coal × 9.5 pounds water)	38,300
Pounds of coal per knot per hour	1,800
Tons of coal per day (24 hours)	410

From the performance curves of the *Delaware* on contract trials it is found she burned 1.85 pounds of coal per I.H.P. per hour at 21 knots. Taking this figure, and using the

same water consumption for auxiliary purposes as that tabulated for 21.56 knots, we get a total expenditure of 518 tons of coal per day.

From the performance curve as calculated by Mr. Emmett for the straight turbine ship, and allowing auxiliary rate same as electric battleship, we get a total expenditure of 503 tons per day.

At 12 knots, using tabulated results from *Delaware* and estimated performances of the turbine ship and the electric ship, we get for total expenditure per day: *Delaware*, 110 tons coal; turbine ship, 104 tons; and electric ship, 87.5 tons. In this calculation the auxiliary water consumption for the turbine battleship was approximated from performances of the *Florida* and *Wyoming*.

The water consumption of the electric battleship for auxiliaries at 12 knots is estimated to be about 25,000 pounds per hour, a little more than that of the *Delaware*, but considerably less than that of the turbine ship, this being due to using only one turbo-generator for the four shaft motors, thus allowing one air pump and circulating pump to be shut down and thereby saving the expense of their operation. It is probable that the water rate for auxiliaries may be even less than that for the *Delaware* as the auxiliaries in operation will not only be less in number, but they will be operating at nearer full load, which is better than running a larger number of machines at a small load.

This principle predominates throughout the turbo-electric installation that the machines in operation are so adjusted that they are generally operating at very near their full load, which, of course, is a more economical condition than where a big machine is required to run at greatly reduced load, such as is the case in the turbine ships at cruising speeds, when the cruising turbine with reduction gear is used and when the steam from this turbine is exhausted into the big H.P. ahead turbine, which will then operate with a load and steam pressure so greatly reduced from those for which designed, it will probably not be efficient and economical. Using the above data, the following comparison is tabulated:

At 21 Knots.

Ship.	Pounds coal per knot per hour.	Tons of coal per 24 hours.
Delaware	2,300	518
Turbine battleship	2,249	503
Electric battleship	1,820	410

At 12 Knots.

Delaware	855	110
Turbine battleship	808	104
Electric battleship	650	87.5

To substantiate the above estimates in regard to the turbine battleship the following data in regard to the French battleship *Jean Bart* is quoted from the "Naval Institute," of September, 1913, pages 1325–6–7. The *Jean Bart* has the following characteristics:

Length on water line, feet and inches	541–4
Beam, feet and inches	88–6
Maximum draught, feet and inches	29–6
Normal displacement, tons	23,467
Designed horsepower	28,000
Designed speed, knots	21
Type of machinery	Parsons turbines, direct-connected.

The turbine ship above referred to has the following characteristics:

Displacement, tons	31,000
Designed S.H.P	31,700
Speed, knots	21
Type of machinery	Curtis turbines, direct-connected, with mechanical reduction gear for cruising turbine.

From the characteristics it is safe to estimate that the E.H.P. of the two ships is about the same. The following are results of official trials of the *Jean Bart*:

Date, April 29, 1913.

Trial	Ten hours, full speed.
Average speed, knots	21.09
Coal per mile, pounds	2,449
Tons coal per day	553.2

Date, May 15, 1913.

Trial	Fuel consumption, cruising speed.
Average speed, knots	12.81
Coal per mile (actual), pounds	937
Tons of coal per day	128

It is stated that the machinery of the *Jean Bart* went through the steaming tests without a hitch, and that for three hours she averaged a speed of 22.04 knots with a run on measured mile 22.63 knots. From this it is seen that her machinery must be highly efficient and satisfactory within the limits of its design.

These results show such an enormous difference in favor of the electric drive that they will undoubtedly be seriously questioned, but the General Electric Company stand ready to guarantee the results within five per cent., and to support the guarantee, as above mentioned, by any reasonable sum of money, and by agreeing to remove the machinery if it does not fulfill the guaranteed performances. As the sum of money is so large, the company do not take this action without being sure of their ground.

The reason that they can be so sure of guaranteed results is that the machinery has all been built, tested and tried out in various conditions on shore, so that the efficiencies and methods of control are accurately determined. The only new engineering involved in electric propulsion for a battleship being that of applying well-tried machinery to the conditions on board ship, and this point should be kept well in mind when considering the proposition, and particularly when comparing the electric installation with the mechanical reduction gear, as the latter is still not in a completely developed state *as a machine* for high powers and speeds, although it has so far proven most satisfactory under certain conditions. But even if the mechanical gear becomes entirely satisfactory for any power and speed, the extra turbines will always be necessary for running astern.

By referring to Plate No. 2, attached, we can see where a following of the mechanical gear will lead, for high powers. This is the plan outlined by Sir Charles Parsons. It contains 22 ahead turbines, 10 astern turbines and 6 sets of mechanical gearing. The sketch does not show the piping, but the mass of it may be imagined. Installations for a battleship of 40,000 H.P. with four shafts, 200 r.p.m.; for a cruiser, 30,000

H.P., 300 r.p.m., and for a destroyer of 20,000 H.P. at 440 r.p.m. were also outlined by Mr. Parsons, all being along the same general plan as shown on Plate No. 2, for battleship of 60,000 H.P. It will be noted that the r.p.m. mentioned for the propellers are much too high for the best efficiency.

The above figures show that the *Delaware* will compare about as favorably with the turbine direct-connected battleship having cruising turbines with reduction gears as she did with the *Utah* and *Florida*, as shown by Captain Dyson.

The *Delaware* was completed in 1909, and it is interesting to note that, if the above calculations are correct, the turbine direct-connected ships, even when fitted with cruising turbines having mechanical reduction gears, are not the equal of a reciprocating-engine ship, considering all the military and economical characteristics desired in a battleship. The trials of the *Texas*, having reciprocating engines, further confirm this.

If the turbine ship mentioned be supplied with reduction gear for each turbine in the manner outlined by Sir Charles Parsons, the steam economy could not equal the electric drive proposed, unless the same turbine and propeller speeds were accomplished. To do this the mechanical gear would have to be built, tested and tried out; in other words, it would be a new design throughout. But even if the same economy were obtained, the difference in weight, complication and, what is of most importance, the maneuvering and backing performances of the ship, would be so greatly in favor of the electric drive that it completely excels the mechanical design.

When considering the care and upkeep of the machinery, it is evident that the electric installation would be superior to all others, because there are no small parts to get out of order; the motion being entirely rotary, the bearings would give no trouble, particularly with forced lubrication; and the turbines being only two in number and therefore very large and of durable construction, the troubles mentiond by Captain Dyson under this heading for turbines direct-connected would largely disappear.

There would be no more reason to expect a large amount of work under care and upkeep of the electric units than is found on machines of this size in control-station power plants, which is practically nothing.

Returning to comparison of direct-connected turbines with other types of propulsion, it was shown by comparison with the *Delaware* that for a battleship the reciprocating engine is even superior to the direct turbine drive. The difficulty of applying the steam turbine to ship propulsion was emphasized in a report made by Rear Admiral Melville and Mr. John H. Macalpine, in May, 1904, and published by Mr. Geo. Westinghouse, 1909, in his book, "Broadening of the Field of the Marine Steam Turbine; the Problem and its Solution." This report was made after an exhaustive investigation of the development of the steam turbine in England and in Europe. In this report the lack of superiority in economy of the turbine over the reciprocating engine, as well as the inferior maneuvering and backing power, was pointed out. Having in mind the latter, it was stated that the Marine-Baumeister, Gustav Berling, of the German Navy, remarked that, "he hoped all the ships of the enemies of Germany would have turbines."

Admiral Oram expressed the opinion "that the turbine would never be suitable for warships on account of the inefficiency at low powers." Sir Charles Parsons recognized the difficulties, and it is interesting to note the enthusiasm with which he has jumped at the mechanical reduction gear.

In concluding the report, it is stated that, "if one could devise a means of reconciling, in a practical manner, the necessary high speed of revolution of the turbine with the comparatively low rate of revolution required by an efficient propeller, the problem would be solved and the turbine would practically wipe out the reciprocating engine for the propulsion of ships. The solution of this problem would be a stroke of genius."

The "solution of this problem" is now undoubtedly accomplished in the mechanical reduction gearing and in the electric-machinery method. Which one to choose depends, as stated

above, on the kind of ship and her desired performances, so that each case must be carefully studied, neither form of speed reduction being regarded as a "sure cure" for all cases.

A study of the requirements of a battleship's propelling machinery shows that the turbine-electric propulsion meets the demands in really a wonderful way; in fact, it is in this type of ship that the electric drive finds its most attractive application, although it is also probable that the development of the submarine cruiser will offer a large field for its use.

Returning to the requirements as given by Captain Dyson it is believed that the above data and comparisons show conclusively that for a battleship the turbine electric-propelling machinery completely outclasses not only the reciprocating engine, but all other types of machinery now developed for the purpose, the points of superiority being as follows:

(1) Economy at moderate and low speeds, hence great cruising radius.

(2) Economy at maximum speed, hence ability to run at highest speeds without exhausting the fuel supply or fireroom force if coal is used.

(3) Great reliability when driven at maximum speed.

(4) Facility of control—high backing power, efficient propellers.

(5) Less weight of machinery and space occupied by it.

(6) Less boiler weight.

(7) Work necessary for care and upkeep, allowing greater radius of duty.

(8) Lack of racing of propellers, hence greater ability to steam at high speeds in a heavy sea.

The last item (No. 8) is accounted for by the fact that the motors are of the synchronous type, and always run at a speed which bears a definite relation to the speed of the generator, which, being direct-connected to the turbine, runs at the turbine speed. As the load comes off the propeller, due to the ship's pitching, the turbine is automatically slowed down by its governor, the latter being of the well-known dependable type used with all turbine installations with certain particular features.

After an earnest consideration of all the engineering features involved in battleship propulsion, the writer has failed to find a single one against the turbine electric drive for this purpose; in fact, all the desirable ones are in its favor, and show the wisdom of the Bureau of Steam Engineering and the Navy Department in making the experiment on the *Jupiter*.

The question now arises, "Are there any reasons why the electric drive shall not be placed on one of our warships of the first class without further delay?" The writer has failed to find any.

It may be feared that great danger from short-circuits, electric burn-outs, etc., may exist, that the electric units might "burn out and blow up." Years of experience on shore with electric apparatus of similar character, power and voltage have shown an almost complete immunity from insulation or short-circuit trouble. In fact, the probability of trouble of this nature on board ship would be less than on shore, as in the ship installation only the amount of electricity required is generated, this being automatically regulated by the turbine governor as the load varies, and there are no possibilities of overload and resulting burn-outs as might occur on shore. Also, the circuits are short and direct, and thus avoid all vibrating strains on the insulation, which is frequently the chief cause of break-down on long circuits carrying high voltage.

The dangers to result from water entering the engine room are no greater than in one with any turbine installation without this machinery; they are less, in fact, as short-circuits can be entirely guarded against and, due to the fact that the four motors can be operated by one turbo-generator, the ship could continue to run at 80 per cent. of her designed speed as long as one of the two turbo-generators rooms were free of excessive water. The windings of these motors are heavily covered by a completely water-proof installation, and if the terminals were properly covered, they could run under water indefinitely. It must be remembered that A. C. current machinery is not subject to the same effects from water as

those using D. C. current. As all the circuits of an induction motor are stationary, it is easy to apply waterproof insulation.

The writer has seen an induction motor (squirrel-cage type) run without difficulty completely submerged in water, and this with no special precautions except to see that the connections were insulated. Of course, the motor is overloaded in this condition, due to the mechanical action of the armature burning against the resistance of the water; but as the water serves as a good cooling agent, the motor can carry a considerable overload in this way without dangerous heating.

Therefore, in case of water entering the engine room, nothing would burn out or blow up. The motors would continue to run until the overload due to the resistance of the water to the revolutions of the motor and the leakage of current became too great for the generator to pull. The point to be understood here is that the electric units would not be the first to be adversely affected by the presence of water. The steam turbine would be in more danger from water than the electric machines, as, if the turbines became partly submerged by water, there would be so much condensation of steam that the turbine could not run. The engine-room steam auxiliary machinery would also be more seriously affected than the motors. So that the presence of the electric machinery in the engine rooms does not cause a more unsatisfactory condition to exist in case the engine room is flooded with water than exists with other types of propelling machinery; in fact, an advantage is gained with the electric installation, because of the fact that the four propellers can be driven by one turbo-generator. Also, the water can be more easily isolated, due to the division of the engine-room space into four watertight compartments which are more independent of each other as shown than they are with other types of machinery.

Reference to Plate No. 2 will show the machinery placed in four watertight compartments so that the flooded compartment can be easily isolated and the ship could continue to run until *both* turbo-generator rooms were flooded.

The danger of personnel would be practically nothing, as

the switches would all be enclosed in an oil bath and the conductors heavily insulated, so that a man would have to almost deliberately set about getting an electric shock by placing his hands on the bars behind the switchboard, and even these could be well protected against such foolish action. There would be no mass of complicated wiring; on the other hand, the conductors would be short, direct and heavily insulated.

With the electric drive the revolutions of the propellers can be accurately regulated from the bridge, and the ship sent ahead, stopped or backed, as desired; provision for all these operations would, of course, be made in the engine room also. But all the trouble and uncertainty of communicating quickly between the bridge and engine room would be avoided. The troubles of voice-tube and telephone communication between bridge and engine room are well known. This was illustrated on one ship recently, when the captain asked the engine room by voice tube if the senior engineer officer were in the engine room; the reply was, "Making 73 (r.p.m.), Sor!"

One thing against the adoption of the electric drive for battleships is that it "sounds too good," but when it is remembered that a thoroughly reliable company is standing ready to put up any sum of "good, hard money," this idea should be forgotten.

Another idea is that if the electric scheme is so good, why do not the merchant ships builders jump at it? The answer is, as pointed out above, the advantage to be gained in ships of constant speed and where maneuvering and control features are not paramount, is not so great as in a battleship.

Also, a largely-influencing factor in this matter is that the shipbuilding companies are not equipped to build in their own shops the electric machinery, and it is therefore against their interest to advocate a battleship having the electric machinery, as it would cause a loss of about a quarter of a million dollars worth of work from the shop of the shipbuilder having the contract.

The opinion has been expressed (see London "Engineer-

ing," December, 1911) that although everything seems to indicate the success of electric propulsion for a battleship, and that it is in a battleship that this method of propulsion finds its best adaptation, yet this very fact prevents a practical demonstration of the design, as it would not be wise to use one of the Navy's capital fighting ships for such an experiment.

It has also been stated that although the General Electric Company would remove the machinery, in case of not fulfilling the guarantees, without money loss to the Government, yet the delay in such a contingency in the final completion of the ship would be too great to warrant taking the chance, even though it might be a remote one. All this, it seems to the writer, depends entirely upon the engineering facts in the matter, as shown by a careful analysis, not a general opinion. If the engineering experience with the machinery to be installed and the data collected, therefore, show conclusively that the machinery will operate as intended, then its application to the purpose of battleship propulsion partakes very *little* of the nature of an experiment, although it would be something new and different from what has heretofore been done; but new departures in engineering of greater magnitude and with less chance of success are being undertaken all the time, and it is in this way real progress is made.

In this connection it should be remembered that there is really more chance being taken with the large marine turbines direct-connected than with the turbine-electric equipment, because, in the former case, a new and untried machine has to be constructed, which, due to the small number in use, has not had opportunity to be standardized; while with the turbine-electric equipment a machine would be used which has been fully developed in its service in land installations. This is evidenced by the well-known difficulties encountered on board ships with the turbine direct-connected machinery as mentioned in Captain Dyson's paper, while little is heard of such troubles in shore-power stations.

The nature of the electric design is such that the entire

propelling machinery could be tested out under all operating conditions in the factory before being placed in the ship. As the General Electric Company would guarantee to build and test the machinery within a year after the receipt of order for it, all defects could be remedied and the machinery would be ready long before the ship would be ready for its installation, and, in the very remote event of the machinery not fulfilling the guarantees on test, machinery of another design could be built for this ship without delaying her date of completion an excessive amount. The chance of any delay of this sort, however, is so small, and the advantages to be gained are so great, that the taking of what chance of this nature that may be thought to exist would be more than warranted, because the engineering facts and experience already obtained from operation of this type of machinery show that it would operate in the ship as intended.

It is believed that the above comparisons show that, when considering the two main characteristics of the propelling machinery of a battleship—namely, those military and those economical—the reciprocating engine as now installed is and will continue to be superior to the turbine direct-connected, even when reduction gears are used with the cruising turbines, and that, on the other hand, the turbine-electric machinery is greatly superior to the reciprocating engine, and that this superiority is so great that its adoption will represent a distinct epoch in the engineering military efficiency of the Navy.

It is desired to again emphasize the importance of the lack of "racing" of the propellers in a heavy sea when driven by the electric machinery. This would enable the ship to run at top speed in any sea, and this fact alone might be the deciding factor in a naval campaign.

Since the above article was written the *Jupiter* has been finished and accepted, her performances exceeding the requirements, and a year's service has demonstrated the reliability and practicability of her electric machinery.

MOTOR-CYLINDER LUBRICATION.

A STUDY OF THE CONDITIONS UNDER WHICH THE LUBRICATION TAKES PLACE, AND OF THE CHARACTERISTICS OF MOTOR-CYLINDER OILS THAT DETERMINE THEIR SUITABILITY FOR THESE CONDITIONS.

By LIEUTENANT G. S. BRYAN, U. S. NAVY, MEMBER.

If an internal-combustion engine, using a certain brand of lubricating oil, runs indefinitely without giving trouble of any kind, we can feel certain of two things; first, that it is a well designed engine, and second, that it is using a good lubricating oil. Otherwise, trouble would occur.

If trouble does occur in an engine, however, we are confronted with several possibilities:

1st. The engine and lubrication system may be poorly designed.

2d. The oil may be poorly refined.

3d. The oil may not be supplied to the cylinders in the right amount.

4th. The oil may be of good quality, but may not be suited to that particular engine.

Trouble due to the first case is exceptional. That due to the second case is more frequent, perhaps, but is also exceptional among well-known brands of oil. That under the third case depends sometimes on the design of the engine, but more often on the lack of care exercised in maintaining the proper oil level in the crank case. The remedy for both of these faults is obvious. Trouble encountered under this head may well depend, however, on the fourth case, in that it is necessary to use an oil that is *suitable* as well as of good quality, or the correct amount will not be supplied the cylinders.

We can assume that most of the engines on the market are correctly designed. From the fact that most of the different varieties of oils on the market work satisfactorily in some engines, we can also safely assume that they are good oils when properly used in the right place. A consideration of the properties of motor-cylinder oils, and an analysis of the conditions under which they lubricate, lead us to the belief that a large majority of the complaints about motor-cylinder oils can be accounted for as due to ignorance regarding the principles governing their use.

Most gasoline motors have their cylinders lubricated by the splash system, the bearings being sometimes lubricated by forced feed. In some cases the cylinders are also lubricated by forced feed, the oil being injected through small holes in the side of the cylinder that are continually covered by the piston. The general principles governing cylinder lubrication, however, apply to both systems.

In the splash system on four-cycle motors the reserve oil is contained in the crank case and each cylinder is lubricated by the splash of its own crank. A small dipper on the end of the connecting rod dips into a trough of oil when the crank is at its lowest point, and with the upward stroke of the crank throws the adhering oil violently against the cylinder walls, splashing it in every direction. This oil is then spread evenly on the cylinder walls by the piston as it slides up and down. Every up-stroke of the crank thus gives a new supply of oil, and every up-stroke of the piston smears some of this oil on the upper part of the cylinder where the combustion takes place.

The operation in a four-cycle motor then is as follows:

Suction stroke.—On the previous up-stroke of the piston oil was spread on the cylinder walls, and this furnishes lubrication as the piston descends on the suction stroke. At the bottom of the stroke the dipper dips into the oil.

Compression stroke.—The film that has just lubricated the piston on the suction stroke will not be affected by the cool gases entering from the carburetor, and will be left intact to

lubricate on the compression stroke. As the piston goes upward it smears a fresh film on the walls. A new supply of oil is also splashed on the lower part of the cylinder by the upward throw of the crank.

Explosion stroke.—The film smeared on the walls on the compression stroke is also sufficient to lubricate the piston on the down stroke. The combustion space on this stroke is exposed to a very high temperature, but the flame only comes into contact with the oil film after the latter has performed its allotted function of lubricating the piston; therefore no difficulty should be encountered. At the bottom of the stroke the dipper again dips into the oil.

Exhaust stroke.—This is the part of the cycle where the most difficulty is encountered with the lubrication. The oil film on the upper part of the cylinder walls has just been exposed to the very high temperature of the ignited gases, and has certainly been damaged to some extent. It still possesses some lubricating value, however, and also the piston itself is now covered with a film which it is about to smear on the walls as it moves upward. The combination of these two factors should be sufficient for the lubrication of this stroke. If trouble is encountered due to faulty lubrication, however, it will show principally on this stroke.

In the two-cycle type conditions are more severe. The oil film is here exposed to the temperature of the explosion every revolution instead of every other revolution as in the four-cycle type. To offset this, however, we have a lower compression and generally slower piston speed. When this is taken into consideration the problem does not appear to be so difficult, and experience has shown that this type can be lubricated efficiently without any particular difficulty.

The three essential requirements of a good motor-cylinder oil are:

1st. It must lubricate the piston efficiently at the temperatures encountered in the cylinder.

2d. It must give a good seal to the piston and rings, keeping them tight and preventing leakage of the oil and condensed gasoline past them.

3d. It must burn without forming carbon deposits *in the cylinder* when an excess of the oil gets into the combustion space.

The first requirement is a general one. If we examine closely into the action of the film of oil we shall see that it is subjected to widely different conditions at different depths. The lubricating value of an oil is due to the fact that when the surfaces of the cylinder and piston are separated by a film of oil the friction of these two metal surfaces is eliminated, and the friction that remains is only that occasioned by the sliding action of the molecules of oil past each other. In other words, it is the same as if two layers of oil are sliding past each other, and the friction that occurs takes place *inside the oil film* and not on the surface between the film and the metal.

The quality of an oil that gives the comparative measure of the friction generated is the *viscosity*, and this may be defined as its resistance to flow. By oil manufacturers it is generally expressed numerically compared to that of water taken as unity; but as known by the consumer it is only expressed roughly as light, medium, heavy, etc., terms that are more generally understood. The higher the viscosity the less fluid the oil, and the greater the corresponding liquid friction. The viscosity of an oil decreases greatly with an increase of temperature, and at the working temperature in the cylinders even the heavy motor oils have their viscosities reduced to about that of a turbine oil at its working temperature.

Figure 1 will show this graphically. Oils A, B and C represent the heavy, medium and light grades of a well-known brand of Pennsylvania (paraffine base) oil. Oil D represents a heavy* oil of asphalt base. It will be noted that the falling off in viscosity with increased temperature is much greater in the case of the heavier grades. Also the viscosities of the Pennsylvania oils are maintained slightly better with increased temperatures.

* The viscosities of the light, medium and heavy grades vary with the different brands. Two "medium" oils, of different brands, for instance, may differ greatly in viscosity.

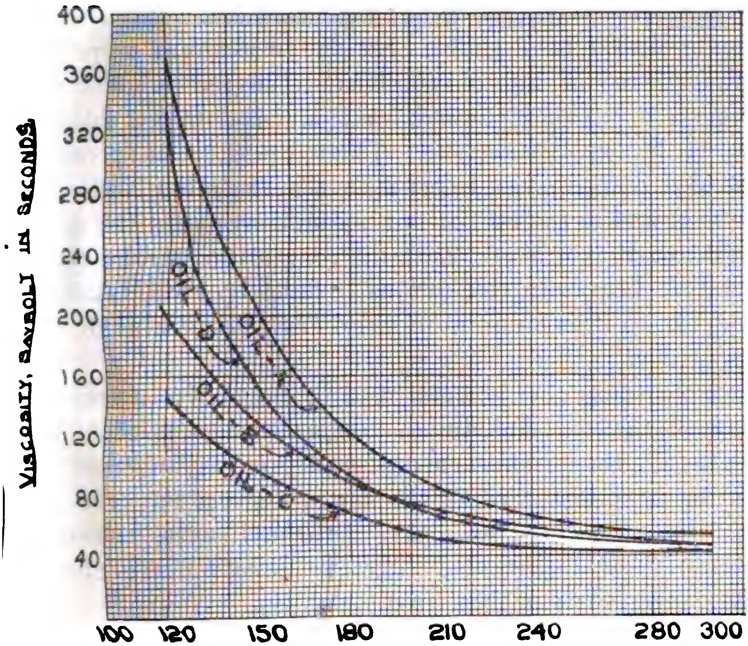


FIG. 1.

The deciding factor with regard to viscosity is the condition of the piston and rings. If these are tight, a light oil will give good results. If they are badly worn and loose, a heavy oil becomes necessary. If the oil is too light, too much of it will be drawn past the piston rings on the suction stroke. Likewise on the compression stroke some of the gaseous mixture from the carburetor will leak past the piston rings and condense in the crank case. As the gasoline which condenses in this way will tend to make the oil still lighter, this effect will be cumulative. Under the conditions of high rubbing speed and small piston-ring pressure that are generally found in motor cylinders, light oils, such as are used in turbines, would probably give the most efficient results if it were not for the high temperatures encountered in the cylinders.

Before discussing the effect of these high temperatures, it

will be well to determine just what they are. Recent experiments abroad have shown that the maximum temperature attained in an internal-combustion engine is about 2,700 degrees F. This is the maximum and is obtained only at the top of the explosion stroke. The temperature is successively lowered as the gases expand on the explosion and exhaust strokes, and as the cool carburetor mixture enters the cylinder on the suction stroke. With the compression stroke the temperature begins to rise gradually until ignition takes place and causes it to jump up to 2,700 degrees again at the top of the stroke.

A recording thermometer used in the above investigations showed that the maximum temperature was about 2,700 degrees F., the minimum about 250 degrees F., and the average during a complete cycle about 950 degrees F. These temperatures are those of gases in the cylinders, and are not those of the cylinder walls. There is a great difference between the temperatures of these two.

An elaborate investigation was carried out by the Bureau of Mines in 1912 * on the transmission of heat in steam boilers, and the successive drops in temperature due to the different media were determined. With the hot moving gases in the boiler at an average temperature of 1,550 degrees F., and the maximum of 2,700 degrees F., and the water in the boiler at 350 degrees F., the following results were obtained :

	Temperature drop, in degrees F.	Percentage of total drop.
Drop in temperature between moving gases and dry surface	1,047	87.3
Drop due to layer of soot on tube	65	5.4
Drop due to thickness of metal	13	1.1
Drop due to layer of scale on water side of tube	65	5.4
Drop between scale and water	10	0.8
Total	1,200	100.0

* Bulletin No. 18, of the Bureau of Mines, 1912, "The Transmission of Heat into Steam Boilers."

In an internal-combustion-engine cylinder the conditions are almost identical except that the temperatures are lower, and there is very little scale on the water side of the cylinder. There is no layer of soot if the cylinder is in good condition, but there is a film of oil on the cylinder walls which would have about the same effect. The point which it is desired to emphasize is the great difference in temperature between the moving gases in the cylinder and the cylinder walls, since this has a very important bearing on the lubrication.

Let us assume that the cooling water in the jacket is at a high temperature—say 200 degrees F., just below the boiling point. Using the average temperature found for the moving gases in a complete cycle (950 degrees F.), this gives us a total difference in temperature between these gases and the cooling water of 750 degrees. If we apportion this drop in the same way as for a steam boiler we have:

	Percentage of total drop.	Temperature drop, in degrees F.
Between moving gases and dry surface...........	87.2	654
Drop due to *film* of oil, assuming it to be the same as for layer of soot.............................	5.4	40
	92.6	694
	7.4	

which would give us the temperature of the inner surface of the cylinder walls as 950 — 694 = 256 degrees F.

Perhaps a more convenient way would be to figure this from the other end, that is, to determine the difference in temperature between the inner surface of the cylinder wall and the circulating water in the jacket. Under the same assumptions as above, this would be 7.4 per cent. of the total drop, which would give a difference of 56 degrees F. If we take the temperature of the circulating water as 150 degrees F., we get a difference of 58 degrees, so we can assume that the temperature of the cylinder walls is about 55 to 60 * degrees higher than that of the circulating water. As long as

* Since this was written some actual experiments conducted have given values of only 30 degrees. See "Steam as a By-product of Internal-combustion Engines," by J. B. Merriam, "Practical Engineer," November 15, 1914.

the water is not boiling we know that the temperature of the walls is little, if any, higher than 267 degrees F.

In air-cooled motors the temperature would be somewhat higher due to the less efficient method of cooling the cylinders. This does not mean that it would become too high, however, as it has been repeatedly demonstrated that properly designed air-cooled motors do not heat up excessively. The lubrication of this type of motors is more difficult at any rate.

With Diesel engines the *maximum* temperature is lower than in ordinary gasoline motors. Since ignition is by temperature of compression, the heating effect begins on the compression stroke; and since combustion is at constant pressure, the resulting *average* temperature in a cycle will be slightly higher. As long as these motors are water cooled, however, the temperature cannot exceed 267 degrees F. without the water boiling.

It was mentiond previously that there was a vast difference between the temperature of the hot gases and the cylinder walls. It naturally follows that the inner and outer surfaces of the oil film will be exposed to quite different conditions. The inner surface is exposed to the high temperature of combustion, and without doubt is very greatly damaged thereby. The outer surface is exposed only to the comparatively low temperature of the cylinder walls, and with a film of any appreciable thickness would be protected from the heat due to low conductivity of this film.

We can consider the film as consisting of two layers, the function of one of these being to furnish the lubrication, and of the other being to withstand the destructive action of heat and to protect this lubricating layer. It is well known that an exceedingly thin layer will furnish lubrication, and it is probable that the greater part of the thickness of the film is used up in giving the requisite protective action from the heat. At any rate, the part that we must look to for lubrication is that part having the lowest temperature, which is the very thin layer next to the cylinder walls.

The action of heat on oils is indicated by two properties,

the flash point and the fire point. The *flash point* of an oil is the temperature to which it must be heated in order that the vapors given off will give a slight explosion when a small flame is held immediately over the oil. The *fire point* is the temperature to which the oil must be heated in order that it will take fire and continue burning when a flame is applied. Roughly it is about fifty degrees higher than the flash point.

It is important that the flash point shall be higher than the temperature of the inner surface of the cylinder, otherwise the vapor given off by the oil will prevent it from adhering to the walls. We have seen that with the water boiling in the jackets the temperature of the inner surface of the cylinder walls will be about 267 degrees F. The temperature of the layer of oil that is in immediate contact with the cylinder walls, which is the part that regulates the friction, cannot be much higher than this. I do not know of any motor oils that have a flash point lower than 325 degrees F. If the temperature of the cylinder walls gets up to this temperature in a water-cooled motor there is something radically wrong, and the remedy is not to get another oil of higher flash point, but to locate the trouble and remove it.

It is an old theory that was never founded on solid facts that a high flash point is a necessity in a motor oil or the oil will burn up without giving any lubrication. The point was overlooked that, when we have a maximum temperature of the gases in the cylinder of 2,700 degrees F. and an average temperature of 950 degrees F., an oil with a flash point of 450 degrees F. will offer but little more resistance to burning than one would of 350 degrees F.

Either oil will burn if kept for any length of time in contact with the hot gas. Lubricating oil does not burn very easily or very fast, however, and the time given for it to burn in a motor cylinder is very short. A thin film of oil smeared on a hot (300 degrees) piece of iron or steel will burn for several seconds if ignited. Few motors ever run at less than 120 revolutions per minute, and at this rate the average point of lubricated surface on the cylinder wall would be exposed to

the action of the flame for only one quarter of a second. It is easily seen that there is no danger of all the oil film being burned in that short time, though there is no doubt that some of it is burned, whether the flash point is 300 F. or 500 degrees F. At high speeds the time allowed the oil to burn is so small a fraction of a second that we need not worry at all on this score. It would therefore appear that a flash point of 300 degrees F. would be sufficiently high for almost any water-cooled motor. Air-cooled motors might in some cases require a higher flash point, but 300 degrees F. should be sufficiently high for Diesel engines except in unusual cases.

The third requirement brings up the question of carbon in the cylinders. So much misinformation has been published on this subject that it will be well to look into the conditions resulting in its formation. In the first place, what is ordinarily known as carbon in the cylinders nearly always contains something else in greater or less quantity. Rust and small particles of iron are nearly always found. In automobile motors a large percentage of dust (silica) is generally present, and in marine motors and Diesel engines salt is a common constituent. In a case recently investigated at the Naval Engineering Experiment Station of an oil that was considered unsatisfactory on account of the large amount of carbon formed, a chemical analysis of the oil from the crank case that was supposedly full of carbon gave results about as follows:

	Per cent.
Free oil,	15
Water,	12
Rust,	11
Salt from sea water,	58
Decomposed oil,	2
Carbon,	1
Foreign matter,	1

Carbon may exist in a motor oil in two forms: First, as free carbon held in suspension, and, second, in combination with hydrogen forming the numerous hydrocarbon compounds

which go to make up the oil. The amount of free carbon in a well-refined oil is very small, and the objectionable carbon deposit is generally due to some other factor.

The conditions attained in the cylinders of internal-combustion engines that result in the formation of carbon are: First, high temperature, and, second, a limited supply of oxygen (air). References have been previously made in this article to the oil "burning." This term has been used rather loosely, as, strictly speaking, "burning" means the combining of the vapors from the oil with the oxygen of the air, and does not include simple vaporization of the oil. Unless air is present in excess of that required for the combustion of the gasoline or fuel oil, and usually it is not, the oil cannot really burn. Under the intense heat, however, the inner surface of the oil film will be vigorously affected, and, in the absence of the air necessary for burning, three things may happen:

Case 1. The compounds may volatilize without decomposition.

Case 2. The compounds may decompose with the formation of free *carbon* and hydrogen.

Case 3. The compounds may decompose with the formation of other hydrocarbon compounds of a different nature.

The products formed in case 1 give no trouble, as, being gaseous, they are carried out with the exhaust, whether burned or not. Of the products formed under case 2, the hydrogen would pass out of the exhaust, whether burned or not. The carbon may be blown out with the gases, or may remain in the cylinder. Whether or not it remained in the cylinder would depend greatly on the condition of the oil film on the cylinder walls. Some oils form a thick, viscous, gummy, asphalt-like deposit, which retains the carbon formed on its surface and prevents it from being blown out through the exhaust. This gummy deposit gradually gets thicker and harder, eventually forming the hard "carbon deposit" so well known in cylinders.

This gummy deposit is due to the action of the compounds mentioned in case 3. The free carbon liberated in case 2 is light and fluffy, and of itself would not form the hard deposit.

Where the compounds break up into new compounds, however, some of the new compounds are volatile, while others are heavier and more viscous than the original compound. Continued action of the kind mentioned in case 3 will therefore result in the gradual thickening of the film; and the retention and absorption by the film of the carbon that is liberated will increase this effect until, finally, a hard, brittle deposit results.

In the absence of any gummy deposit of this kind to cement the free carbon together, the latter will generally be blown out through the exhaust. The oil that will give the best results, then, is not necessarily the one that will *form* the least carbon, but the one that will *form the least carbon in the cylinders.*

Oils made from the Southern-asphalt-base crudes have shown themselves to be much better adapted to motor cylinders, as far as their carbon-forming proclivities are concerned, than are the paraffine-base Pennsylvania oils. The carbon formed from the latter is, as a rule, extremely hard and clings to the metal surfaces, while that from the former is soft and can easily be wiped off any surface that it is deposited on. This would be expected from a consideration of the nature of the hydrocarbons composing the oil, and it has also been demonstrated in practice.

The explanation lies in the fact that the paraffine-base oils are generally composed of the paraffine series of hydrocarbons, while the asphalt-base oils are composed mainly of the ethylene and napthene series. One of the characteristics of the latter two series as compared with the paraffine series is their tendency to distill without decomposition. Consequently no gum will be formed on the cylinder walls, and the carbon liberated will be mostly discharged with the exhaust gases.

In the light grades of motor oils there would probably be very little difference between the two varieties, but in order to get oils with a high viscosity in the paraffine brands it is necessary to compound the light oils with cylinder oil in different proportions. The presence of this cylinder oil is what is responsible for most of the gumming, and this explains

why the heavy oils give so much more carbon trouble than the light oils. Oils from an asphalt base can be made with viscosities sufficiently high to make it unnecessary to compound them with cylinder stock.

Some oil companies, in their advertising, lay great stress on the color and specific gravity of their oils. The color of an oil has no relation to its lubricating value. About the only information of any value that can be obtained from noting the color is whether or not it is a light oil compounded with a cylinder oil. If such is the case it shows the familiar greenish tinge of the cylinder oil.

The specific gravity is of no practical value to the consumer. The producer uses it to advantage in his refining. It might be used in conjunction with the flash point to determine what particular kind of crude a straight oil is made from, but it would take almost an expert to do this.

It will generally be found the best policy to use a lighter oil in winter than in the summer. There is generally very little difference in the working temperature of the cylinder walls in winter and summer, but the temperature of the crank case will show very great differences. If the oil is very viscous, due to low temperature, it will not splash very easily and too little oil will be supplied the cylinders. This will be particularly apparent when starting the engine. The *cold point*, which is the temperature at which the oil freezes, should be sufficiently low to insure that no difficulty will be encountered at the temperatures to which the crank case will be exposed. Otherwise it is of little value. The asphalt-base oils have a lower cold point than the paraffine oils. The heavy oils are usually the only ones that have cold points that are not sufficiently low.

The amount of oil necessary for the lubrication of motor cylinders depends on so many factors that no data can be given, and it will be necessary to solve the problem for each individual motor. The heavier the oil the less the amount that will be used; also the tighter the piston and rings the less will be used. It is also well to remember that to use

more than is necessary for proper lubrication simply aids in the formation of carbon.

Two-cycle motors can be successfully lubricated by mixing the lubricating oil with the gasoline in the gasoline tank. The proportion generally used is about one pint of oil to five gallons of gasoline. In this method of lubrication the oil must vaporize in the carburetor and be carried with the gasoline vapor into the cylinder, where it will condense on the walls. In choosing an oil for this method of lubrication attention should therefore be paid to the characteristics governing its vaporization.

Gasoline, kerosene, lubricating oil and most all other products obtained from crude oil consist of a large number of different kinds of hydrocarbons, each of which has its own boiling point. When the oil is heated the lighter hydrocarbons will first vaporize, being followed by the others as t boiling point is reached.

If we take a mixture of gasoline and lubricating oil heat it so as to cause it to vaporize, the gasoline will be dri off and the oil will remain, unless we select such conditi that the oil will also vaporize without leaving any resid In the carburetor conditions are made very favorable vaporization, but it will help appreciably if an oil is selec that has a low flash point and is easily vaporized. The wr has in mind a case that was brought to his attention a few da ago of a motor-boat engine that gave considerable difficul owing to the oil congealing in the carburetor in cold weathe.

The study and application of the foregoing principles to the selection of a motor oil for use in any particular engine should enable anyone to obtain an oil that is suited to that engine. It should also clear up some of the obscurity regarding the reasons why an oil will work well in some engines and will not in others. The writer believes that the formation carbon in the cylinders of internal-combustion engines is evil that can be prevented by a little study of its causes.

TESTS OF MATANUSKA COAL, *U. S. S. MARYLAND.*

From Report of a Board consisting of Captain Philip Andrews, U. S. Navy; Lieutenant Milton S. Davis, U. S. Navy; Lieutenant H. E. Kays, U. S. Navy, and Ensign W. O. Henry, U. S. Navy.

ORDER OF TESTS.

(1) *Seven-day port test;* 4:00 A. M., November 7th, to 4:00 A. M., November 14th, 1914; test held at the Navy Yard, Puget Sound, Wash.

(2) *Four-hour full-power test;* 1:30 P. M. to 5:30 P. M., November 14th, 1914; test held in Puget Sound and Straits of Juan de Fuca.

(3) *Twenty-four-hour fifteen-knot test;* three-quarter boiler power; 10:45 A. M., November 15th, to 10:45 A. M., November 16th, 1914; test held in the Straits of Juan de Fuca and Pacific Ocean en route to Mare Island, California.

(4) *Forty-eight-hour ten-knot test;* 12 noon, November 16th, to 4:00 A. M., November 18th, 1914; test held in the Pacific Ocean making passage to Mare Island, California.

SEVEN–DAY PORT TEST.

Boiler No. 8 was used during the entire week. The usual auxiliaries were run; evaporators were used when necessary for filling the ship's tanks. The load on the ship was rather heavy; somewhat heavier than with that during the tests of Pocahontas or Bering River coals. It will be seen by attached data sheets that the load is proportionate to the total water evaporated. The water evaporated was measured as in previous tests.

The port auxiliary machinery and condenser were used. All steam drains were led into the port feed tank. Water was pumped across to the starboard feed tank, where it was measured before pumping into the boilers. All coal, ash and clinker were weighed; during day watches of two days the load was so excessive for one boiler that it became necessary to assist the draft by running the blowers. Run-of-mine coal was used for five days, slack for one day and lump coal for part of one day. There were no casualties. Data was very carefully collected by the assistant engineer officers, who stood a strict watch during the test. At the conclusion of each watch the assistant engineer officer submitted a report of data for his watch on forms which had been previously prepared. During this test the coal burned very freely. The firing was very good, the analyses of the flue gases giving rarely below 9 per cent. of CO_2. There was little clinker, but the ash was several per cent. higher than with the Pocahontas coal. The firing was carefully supervised, the officer on watch and chief water tender being present at all times. The draft was good and the coal burned with bright, yellowish flames. The coal coked very nicely; the coke was friable and very easily worked by the firemen. Fires six to eight inches thick were carried most of the time, though occasionally they were heavier. Two fires were cleaned each watch, although they might have been operated longer. The ash fused into clinker on the grate bars, generally about two inches thick, medium weight, porous, a little tough and hard while hot, but friable when cold. The clinker had a little ash mixed throughout the mass. It stuck a little to the bridge wall, but not seriously. The soot deposit was about 25 per cent. more than with Pocahontas coal. The soot was a little different from that of the Pocahontas, as the granules appeared as minute fused grains.

The work of firing this coal was easy for the firemen. Hoes were used for spreading the fires, which were kept level. The slice bar was used sparingly. The load of the ship was carried with ease except when coaling ship or cleaning up

after coaling, when there was a heavy drag on the boiler and the draft had to be assisted by the blowers. The load during this week of test would have ordinarily required two boilers burning Pocahontas coal.

FOUR-HOUR FULL-POWER TEST, FORCED DRAFT.

The *Maryland* stood by to get underway at 8:30 A. M. with all boilers, but, owing to the fog, she was unable to get underway until 11:19 A. M. The speed was gradually worked up to 120 turns about 1:30 P. M., when the test commenced. The fires had been worked and the ship made 108 turns under natural draft when the blowers were started and a draft of about one inch of air pressure in all firerooms obtained. The speed of 119 to 120 turns was very easily made. The fires were carried thin; blowers regulated to about 350 revolutions; dampers were partly shut. After all the fires were in good condition it was possible to regulate the steam pressure by operating the ash-pan doors. A very uniform steam pressure was carried, as will be seen in the attached data sheet of gage pressures. The fires burned brightly, the work of firing being very easy on account of the ease of breaking up the coke. The furnaces were one mass of yellowish flame. There was not an excessive amount of ash formed. The men, on being questioned, all said that it was the easiest twenty-knot run that they had ever made.

TWENTY-FOUR-HOUR FIFTEEN-KNOT TEST.

The *Maryland* anchored in Port Angeles after the four-hour forced-draft test. During the evening and following morning the usual cleaning work was done on all boilers; ash pans and furnaces were hauled and all ash and clinker separated and weighed. Tubes were blown with steam and all soot boxes hauled. The furnaces were primed with Matanuska coal. Boilers Nos. 4, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16 were lighted up. At 10:00 A. M. the *Maryland* got underway and stood to sea, setting a course for San Francisco

Bay. By 10:45 A. M. the speed had reached 85 turns, which was the speed set to make 15 knots, and the test began. Data was collected as per sample forms. It soon became evident that the boiler power was much too great for the speed if the fires were to be properly worked in order to get the highest efficiency. Boilers No. 4, 6, 15 were first banked and then allowed to die out. No. 16 boiler was banked and the speed maintained with eight boilers. The firing was by signal, the intervals being about nine minutes. The same general remarks apply to the burning of the coal on this test. It burned with the greatest of ease, forming a very easily-worked coke, not an excessive amount of ash or clinker and, in general, appeared to be easier to handle than the Pocahontas coal. The CO_2 found in flue analyses was generally high, over 9 per cent. Fires, carried about six to eight inches thick, were always glowing, ash pans bright and the furnaces a mass of yellowish incandescent flame. The coal burned like pine knots. The amount of clinker was not excessive, was more or less easy to work and was very friable when cold. The fires were noticeably hot. The amount of soot made was a little higher than is generally made by good Pocahontas coal, about 10 per cent. more.

TEN-KNOT TEST.

This test was commenced at 12 noon, November 16th, the fifteen-knot test having been finished at 10:45. Ash pans and furnaces were hauled; tubes blown and soot boxes hauled; all fires were cleaned and were built up. Boilers Nos. 9, 10, 11, 12, 13, 14 were used, but, as in the fifteen-knot test, it was found that there were too many boilers for working fires properly. Two boilers were ultimately cut out and the test made with four boilers, which were more than ample, as, at times, there were four evaporators in use. The same remarks as to the fires on the fifteen-knot test apply to this test. Fires were very easily worked; the coke broke up easy; clinker was not very hard; bright, level fires from six to eight inches thick and sometimes thicker were carried, and the CO_2 was

generally high. After running forty hours it was found there were but three tons of Matanuska coal left, so the test was considered completed after forty hours.

GENERAL.

The Matanuska coal was delivered by the steamship company to the coaling plant at the Navy Yard, Puget Sound, where it was stowed under cover. At the time of the arrival of the *Maryland* it was still sacked. Some of the sacks had became rotten and a little coal had been spilled on the floor of the dock. The coal was inspected by the officers of the Board and the assistants in the engineering department. It was found to be very slack, as will be seen in the data sheet; also, it was very dry. There were very few lumps larger than a man's fist. A large working party under Ensign Bowden was sent to the coal sheds to load the coal on lighters. Every bag was weighed and spread out for inspection. All extraneous matter observed was picked out. There was very little slate, not over two hundred pounds being picked out of the whole lot of 586 tons. About forty tons was laid aside and screened into two sizes, that which would go through a quarter-inch mesh and that which would not. Of the former there was about twenty-five tons and of the latter about fifteen tons. This coal was burned on successive days, one day being devoted to the slack and a part of another day to the lump. The slack appeared to burn better than the lump, and the reason will be seen in the chemical-analyses sheet attached. When the lighters were filled they were towed alongside the ship and the coal was stowed in bunkers which had been previously swept clean. During the test there was no evidence of any gases being given off the coal. During the loading, in order to avoid dust, the coal was thoroughly wet down, but even with this precaution it was very dusty. The noticeable characteristic of this coal is its friability. Lumps pulverize very easily. This accounts for the chemical analysis, which shows the slack to contain less extraneous matter than the

lump. It will be seen that the coal, being friable, powders very easily, while the lump contains more or less shale and is hard. In this respect the coal shows a marked difference from the Bering River coal, where the lumps contained a small per cent. of extraneous matter while the slack contained about 30 per cent. Before loading the lighters samples at random were taken from the pile. These were sized into slack, buckwheat and lump, and were taken to the chemist for analysis as to ash. These samples contained from six to ten per cent. of ash and, on account of this showing it was decided that nothing further than hand cleaning would be necessary.

PHYSICAL AND CHEMICAL CHARACTERISTICS.

Moisture: There was apparently no free moisture on inspection before loading. This coal was stowed in a dry place and was extremely dry and dusty. Density 40.3 cubic feet per ton alongside and 39.8 cubic feet per ton at coal dock.

Sample: The laboratory was obtained by taking a shovelful out of every fifth bag as it came on board the lighter. This was bagged and stowed in a dry place. There were about fifty bags in this pile. On the day the laboratory samples were taken these bags were dumped out and the pile thoroughly mixed. As this pile was about five tons in weight, it was reduced and every fifth shovelful saved. It was again quartered and then quartered again. This gave a 325-pound sample, which was taken on board ship and then quartered and pulverized until there was a proper amount for chemical analyses and shipment to the Bureau of Steam Engineering and the Bureau of Mines.

General sample: Results of analysis are as follows:

Moisture, per cent	0.73
On dry basis:	
Volatile combustible matter, per cent	20.29
Fixed carbon, per cent	69.32
Ash, per cent	10.39
Sulphur, per cent	0.49

	Slack.	Buck-wheat.	Lump.
Moisture, per cent	0.75	0.65	0.63
On dry basis:			
Volatile combustible matter, per cent	20.96	20.22	19.47
Fixed carbon, per cent	70.85	70.50	65.19
Ash, per cent	8.19	9.28	15.34

Analyses by Mr. North, Yard Chemist, Navy Yard, Puget Sound.

A sizing sample of 785 pounds was obtained which gave the following:

Lump which would not go through a one-half-inch mesh screen, 17.7 per cent., 139 pounds.

Buckwheat which would not go through a one-quarter-inch mesh screen, 12.1 per cent., 196 pounds.

Slack which would go through a one-quarter-inch mesh screen, 70.2 per cent., 551 pounds.

Slate: There were no impurities in this coal of any consequence. Little or no slate could be found.

Calorific value: This could not be obtained.

Smoke: This coal makes some smoke, considerably more than Bering River, and about the same amount as Pocahontas.

Indicator cards were taken once an hour during the full-power test and once a watch during the other tests. The horsepower found agreed with the curves of the horsepower on board for the twenty-knot test, but the fifteen-knot test and ten-knot test appeared to be somewhat under that generally found for that of fifteen and ten knots. The indicators are old and the results obtained were irregular.

CONCLUSION OF TEST.

At the conclusion of each test an inspection of the boilers was made. No warping or overheating occurred during the four-hour full-power run. During the fifteen knot test the uptakes of No. 7 boiler were warped a little. In the uptake of No. 13 boiler there were ten rivets pulled out and the uptake door sprung a little. A few panel doors on the boilers show signs of some of the insulation being burnt out. From

the above it will be seen that this coal has no bad effects on the boilers or uptakes. There were no other casualties.

There were seven grate bars renewed during the port test.

The work of this test has been entered into by the officers and men concerned with the greatest enthusiasm. No effort has been spared to obtain the best results. All other ship work has been made secondary to the coal test. The machinery and boilers of the *Maryland* functioned perfectly during all these tests.

Mr. S. B. Flagg, Engineer of the Bureau of Mines, was present in the firerooms during these tests. He did much to assist the ship's force in working fires and instructing the firemen in the best methods of obtaining efficient combustion of this coal. The Board is also indebted to Mr. Flagg for advice and assistance in compiling the data of this report.

COAL ACCOUNT.

Receipts.

(*a*) By invoice, tons	586.0
(*b*) By weight, tons	585.7
(*c*) By tally, tons	583.0
(*d*) By bunker estimate, tons	574.0

Expenditures.

(*a*) Loss in handling (due to fineness), tons	12.000
(*b*) Priming and preparatory to port test, tons	4.000
(*c*) * Port test, tons	116.185
(*d*) Priming and preparatory to 4-hour test, tons	45.000
(*e*) * Four-hour test, tons	85.849
(*f*) Coming to anchor, tons	8.000
(*g*) Priming and preparatory to 15-knot test, tons	19.000
(*h*) * Fifteen-knot test, tons	161.112
(*i*) Preparatory and cleaning fires for 10-knot test, tons	8.000
(*j*) * Ten-knot test, tons	120.582
Total, tons	579.728
Difference, tons	6.272
On hand, about, tons	3.000

Bunker Estimate: Inspection of Bunkers.

Before tests, tons	574.000
After port test, tons	450.000
After four-hour test, tons	328.000
After 15-knot test, tons	120.000
After 10-knot test, tons	3.000

* These quantities by careful tally or actual weight. Other figures by estimates.

After investigating and reconciling all data the Board submits the following tables of comparisons with Pocahontas, Bering River and Matanuska coals.

Chemical comparison of coal.—Samples of which the analyses are given below were taken from Pocahontas and Bering River coals tested by U. S. S. *Maryland* in 1913.

Name.	Moisture.	Volatiles.	Fixed carbon.	Ash.	Sulphur.	B.t.u.
Pocahontas	2.5 2.4	19.2 19.0	74.6 74.6	6.2 6.4	.75 .65	14,660 14,600
Bering River	1.6 1.7	17.7 16.7	61.9 59.9	20.4 23.4	.65 .60	12,180 11,740
	3.5 1.0	16.3 17.5	58.8 69.9	24.9 12.6	.60 .60	11,410 13,434
Matanuska	.73	20.29	69.32	10.39	.49	*

* No apparatus available for making this determination.

Analyses of Pocahontas and Bering River coals supplied by Bureau of Steam Engineering. Samples of Matanuska coal are taken as herein described. Analyses of Matanuska coal by Mr. North, Navy Yard Chemist, Puget Sound.

Densities.

Pocahontas, average, cubic feet per ton 42.16
Bering River, average, cubic feet per ton 39.83
Matanuska, average, cubic feet per ton 40.30

Comparison of Port Consumption.

PORT TEST OF SEVEN DAYS.

Coal.	Total tons.	Gals. water evaporated.	Gal. of water per pound of coal.	Ash, per cent.	Efficiency, per cent.
Pocahontas	94.291	248.610	1.177	11.04	100.
Bering River	136.391	247.783	.811	36.6	68.9
Matanuska	116.185	305.446	1.173	15.8	99.6

FOUR-HOUR FORCED-DRAFT, SPEED, TWENTY KNOTS.

Coal.	Total tons burned.	Percentage of ash.	Smoke by scale.	Knots per ton.	I.H.P.	Pounds per I.H.P.	Steaming radius.	Average efficiency.
Pocahontas...	79.1	8.8	2.4	1.02	20,820.3	2.09	2,367.8	100
Bering River*	127.3	38.8	1.5	.60	13,992.3	5.32	...	...
Matanuska....	85.848	18.67	2.8	.93	19,929.15	2.32	2,000.2	91

* During this test speed of 20 knots could not be obtained.

FIFTEEN-KNOT TEST, TWENTY-FOUR HOURS.

Coal.	Total tons burned.	Percentage of ash.	Smoke by scale.	Knots per ton.	I.H.P.	Pounds per I.H.P.	Steaming radius.	Average efficiency.
Pocahontas...	153.155	7.6	1.25	2.38	7,083.	†2.01	4,781.	100
Bering River..	160.3	35.0	.60	1.09	7,600.	4.98	2,372.	43
Matanuska....	*157.212	14.59	1.99	2.29	6,142.37	2.15	4,796.3	95

* Allowance of three tons of coal used on boilers banked or cut out.
† I.H.P. taken from curve of revolutions and I.H.P., as data obtained from indicator cards irregular.

TEN-KNOT TEST, FORTY-EIGHT HOURS.

Coal.	Total tons burned.	Percentage of ash.	Smoke by scale.	Knots per ton.	I.H.P.	Pounds per I.H.P.	Steaming radius.	Average efficiency.
Pocahontas...	137.325	10.5	1.18	3.515	2,134	†3.08	7,077	100
Matanuska ...	*118.582	15.67	1.86	3.37	2,686.527	3.09	7,160.6	98

* Allowance of two tons for coal used on boilers banked and cut out.
† I.H.P. taken from curve, as data obtained from indicator cards irregular.

Bering River coal expended before this test.

Matanuska coal expended after forty hours.

The Board finds that this sample of Matanuska coal tested is suitable in every respect for use in the naval service.

DESCRIPTION OF THE REPAIR PLANT OF THE U. S. S. *VESTAL.*

BY LIEUTENANT COMMANDER L. J. CONNELLY, U. S. NAVY, MEMBER.

The *Vestal* is a vessel with an overall length of 465 feet 10 inches, extreme breadth of 60 feet 2½ inches, and a normal displacement of 7,720 tons. (When in service as a collier, before conversion as a repair ship, her loaded displacement was 12,585 tons.) She has two triple-expansion, vertical, inverted, direct-acting, three-cylinder engines of 7,500 horsepower, driving two three-bladed propellers. Speed, fourteen knots per hour. Steam being furnished by six B. & W. boilers at 180 pounds working pressure.

The *Vestal's* conversion from a collier to a repair ship was authorized by an act of Congress making appropriations for the Naval Service for the fiscal year ending June 30th, 1913.

The conversion required the installation of decks and galleries in the former cargo holds; the relocation of two of the original four masts, the masts being now so placed as to permit the use of traveling cranes in the shops. The installation of additional electric and pneumatic power and distilling apparatus was necessary. A number of structural changes were made: fresh-water tanks were built in between frames forty-three and forty-seven in order to carry potable water for the larger crew; the superstructure was extended aft to provide for an optical workshop, a general office, and a drafting room; skylights, sixteen feet by fourteen feet, were built to cover the original cargo hatches; six cargo ports, three on each side, were cut so as to give light and ventilation to the work shops and to facilitate the handling of work delivered alongside in boats. A towing engine has been installed aft, so that the ship may take another vessel in tow in case of necessity.

The layout of shops is as follows: No. 1 hold, frames twenty-seven to forty-seven, is now the *General Storekeeper's Store* for repair materials. In this compartment two platform decks and the hold proper are given over entirely to repair stores; over nineteen hundred sheet-steel bins, one foot high, one foot wide and three feet long, have been installed. A thousand or more of the bins have been divided into smaller spaces by sheet-steel division plates. Racks for the stowage of bar metals, pipe, sheet metals, lumber, etc., have been provided, giving, in all, stowage space for about eight thousand items of repair material. A visitor on board remarked that "It is like a three-story hardware store, being three stories deep instead of three stories high." A power hack saw is installed near the bar-metal storage. A computing scale for weighing out quantities of small articles is installed near the bins containing fittings, bolts, etc. The general storekeeper's department is a very important part of the repair ship, it being necessary to carry a working stock to cover all ordinary expenditures while the ship is away from a supply base.

The *Machine Shop* is located between frames forty-seven and sixty-four. The shop has two galleries (one to starboard, one to port) and a lower deck. The heavier tools are located on the lower deck, the lighter tools, in the galleries. A three-ton electric crane travels fore and aft the shop. All the machine tools are driven by independent electric motors. The starboard machine-shop gallery contains the following machine tools:

Two 12-inch × 6-foot lathes, driven by 1½-H.P. variable-speed electric motors.

Two 14-inch × 8-foot lathes, driven by 3-H.P. variable-speed electric motors.

One 16-inch × 10-foot lathe, driven by a 5-H.P. variable-speed electric motor.

One 18-inch × 6-foot lathe, driven by a 7½-H.P. variable-speed electric motor.

The six lathes mentioned above are of the "Prentice" high-

speed, geared-head type, and driven with silent-chain belts by Reliance motors of the 120-volt, semi-enclosed, variable-speed, direct-current type. The motors are placed at the lowest part of the head stock column.

The other tools on this gallery are:

One 2½-inch × 26-inch Acme flat turret lathe, driven by a 3-H.P. Reliance motor.

One Pratt and Whitney ⅝-inch × 4½-inch turret lathe.

One Hendey 4-inch centering machine, driven by a ½-H.P. constant-speed Reliance motor.

One 14-inch × 6-foot American Tool Company's geared-head lathe, driven by a 2-H.P. v. s. Westinghouse motor.

One 30-inch Mueller plain radial drill, driven by a 3-H.P. c. s. Reliance motor.

One 12-inch Willey electric-driven sensitive drill.

One Willey electric wet-tool emery grinder, the motor being contained in the pedestal.

The port machine shop gallery contains the tool cage, in which is kept for issue on check portable pneumatic and electric tools, hand tools, reamers, etcetera, also a Wilmarth and Morman combined cutter, reamer and twist-drill grinder, driven by a 1-H.P. General Electric motor; a Prentice 12-inch × 6-foot toolmakers' lathe, driven by a 1½-H.P. v. s. Reliance motor; and a Gorton engraving machine, driven by a ½-H.P. Reliance motor.

Outside of the tool cage, on the gallery proper, the following machine tools are installed:

One 12-inch × 6-foot Prentice toolmakers' geared-head lathe, driven by a 1½-H.P. variable-speed Reliance motor.

One No. 2-B, heavy, Brown and Sharpe, plain milling machine, with a 7½-H.P. c. s. Reliance motor.

One No. 3-B, Brown and Sharpe, Universal milling machine, with a 7½-H.P. c. s. Reliance motor.

One No. 4-S Cincinnati plain milling machine, with 10-H.P. c. s. Reliance motor.

One 16-inch Willey sensitive drill, electrically driven.

One Riehle 100,000-pound testing machine, driven by a 5-H.P. Crocker-Wheeler motor.

One Cincinnati electric-driven two-wheel bench grinder.

The lower deck of the machine shop contains the following tools:

One 18-inch × 8-foot Prentice geared-head lathe, driven by a 7½-H.P. v. s. Reliance motor.

One 18-inch × 10-foot Prentice geared-head lathe, driven by a 7½-H.P. v. s. Reliance motor.

One 32-inch × 14-foot Prentice geared-head lathe, driven by a 15-H.P. c. s. Reliance motor.

One 25-inch—39-inch × 6-foot—10-foot Harrington extension-bed gap lathe, driven by a 7½-H.P. c. s. Westinghouse motor.

One 48-inch—72-inch × 8-foot—15-foot Harrington extension-bed gap lathe, driven by a 7½-H.P. v. s. Reliance motor.

One 10-inch New Haven Manufacturing Company's slotter, with 3-H.P. c. s. Reliance motor.

One 18-inch × 30-inch × 96-inch Norton plain-cylinder gap grinder, driven by a 15-H.P. c. s. Reliance motor.

One 72-inch Franklin boring, drilling and milling machine, driven by a 10-H.P. c. s. Reliance motor.

One 72-inch J. M. Poole vertical boring mill, with a 10-H.P. v. s. Reliance motor.

Two 25-inch Snyder heavy-duty upright drills, with 2-H.P. v. s. Reliance motors.

One 36-inch × 8-foot Detrick and Harvey open-side planer, with 10-H.P. c. s. Reliance motor drive.

One 30-inch × 30-inch × 72-inch Woodward medium metal planer, driven by a 5-H.P. c. s. Reliance motor.

One 16-inch Flather heavy-duty crank shaper, driven by a 5-H.P. Reliance motor.

One 15-inch Potter & Johnston shaper, driven by a 1-H.P. c. s. General Electric motor.

One 24-inch Flather heavy-duty crank shaper, driven by a 7½-H.P. Reliance motor.

One Willey electric-driven wet-tool emery grinder.
One Willey electric-driven two-wheel emery grinder.
One Royal No. 8 power hack saw, ½-H.P. motor.
One 4-foot Drese's full universal radial drill, driven by a 5-H.P. v. s. Reliance motor.
One buffing lathe, with motor incorporated in the stand.
One 24-inch Bullard vertical turret lathe, driven by a 5-H.P. c. s. Reliance motor.
One Lovekin 7-inch pipe expanding and flanging machine, driven by a 25-H.P. Westinghouse motor.

A gage-testing outfit, vice benches, arbor presses, floor stand with chuck, portable Franklin cranes, portable boring bars for cylinders, and main-engine bearings, shaft-straightening press, and miscellaneous equipment have been provided.

The *Electric Shop* is located underneath the lower deck of the machine shop. It is fitted up with a work bench, armature horses, a Hendey lathe, and a small electric-plating motor generator for nickel plating.

Number three hold contains a coppersmith shop on the starboard gallery, a pattern-making shop on the port gallery, and the lower deck is a combination of blacksmith, shipfitter, and boiler shop.

The *Coppersmith Shop* outfit consists of:

One horizontal hydraulic pipe-bending machine, power being supplied by an electric-driven Gould's triplex power pump.
One 3-inch Stoever pipe machine, driven by a 3-H.P. c. s. Reliance motor.
One Watson-Stillman hand-power, hydraulic, stanchion pipe bender.
One hand-power sheet-metal folder.
One 22-inch upright drill, driven by a 1¾-H.P. c. s. Westinghouse motor.

Bench machines, hand tools, forges, compressed-air fuel-oil torches, and a lead-lining equipment, has been provided.

The *Patternmaking Shop* is located in the port gallery of this compartment and has the following machine-tool equipment:

One electric-driven band-saw filing machine.

One 36-inch × 60-inch × 14-foot Fay and Scott extension-gap patternmakers' lathe, with 4-H.P. Reliance motor.

One 12-inch Oliver motor-head patternmakers' speed lathe, with ½-H.P. Oliver v. s. motor.

One Crescent universal woodworker, driven by a 7½-H.P. c. s. Reliance motor. (This machine has a band saw, circular saw, shaper, jointer, morticer, borer and sand disc.)

One No. 2 Oliver universal wood trimmer.

One Crescent universal double-arbor saw bench, driven by a 5-H.P. c. s. Reliance motor.

Two No. 5 Fox bench wood trimmers.

One Crescent motor-driven hand planer and jointer, with 5-H.P. c. s. Reliance motor.

One 24-inch Wood's motor-driven surfacer, 7-H.P. motor.

One electric-driven oilstone grinder, equipped with the following wheels: one fine and one coarse oilstone wheel, one emery cone, one leather stropping wheel, and one emery wheel. Manufactured by Mummert Dixon Co.

One electric-driven jig saw, 6-inch stroke; motor, 1½-H.P.

One electric-driven friezing and shaping machine; manufactured by J. A. Fay & Co.; driven by a 3½-H.P. Westinghouse motor.

One electric-driven morticing machine manufactured by H. B. Smith Machine Co., with 3-H.P. General Electric motor.

One electric-driven 16-inch × 6-foot patternmakers' lathe.

Bench, tools, electrically-heated aluminum glue pots, saw-brazing equipment, etcetera, have been provided. A number of standard patterns (about 300) are on hand as part of the equipment.

The lower deck of this compartment is a combination of the smithy, ship-fitter, and boiler shops. Four forges, one tempering furnace, bending slabs, the oxy-acetylene cutting and welding apparatus, with usual anvils, work benches, a three-ton traveling crane, and miscellaneous tools, and the following power-driven machines, comprise the equipment:

One 150-lb. steam hammer, driven by compressed air.
One 150-ton high-speed hydraulic forging press. This is one of the Marine type, made by the United Engineering and Foundry Company of Pittsburgh.
One 30-inch Mueller plain radial drill, with 3-H.P. c. s. Reliance motor.
One 28-inch Newton cold-metal sawing machine.
One motor-driven hack saw.
One Wicke's Bros. double-end punch and shears. Punches and shears up to 1 inch, driven by a 7½-H.P. c. s. Reliance motor.
One Cleveland 6-foot geared motor-driven plate-bending roll, 10-H.P. Reliance motor.
One power shears for metal up to ½ inch, with 30-inch throat.
One 30-inch power rotary shears.
One electrically-driven wet-tool grinder.

The forges use low-pressure oil burners, the blast being furnished by a Connersville positive-pressure blower, driven by a 7-H.P. electric motor.

An electrically-heated oil-tempering bath (manufactured by the General Electric Co.), with necessary switchboard control, also a pyroscope and a sclerescope for testing hardness of temper, are provided. A Lincoln arc-welding (motor generator) outfit is installed in the starboard forward corner of this shop.

Number four hold contains the *Foundry,* on the lower deck; the *Coremaking Department,* on the starboard gallery; and the *Carpenter Shop,* on the port gallery.

The *Foundry* has one 1 to 2-ton cupola; one 2½ to 3½-ton cupola; the blast being furnished by a B. F. Sturtevant pressure blower driven by a 17½-H.P. electric motor.

There are four Rockwell tilting-crucible brass furnaces, two with working capacity of 200 pounds, and two with working capacity of 375 pounds, fuel-oil fired; their blast being furnished by a General Electric centrifugal compressor driven by a 10-H.P. General Electric motor.

A six-crucible steel furnace will shortly be installed, using oil and natural draft.

In one corner of the foundry is a small sand-blast room, also a metal-cutting band saw, electrically driven, for cutting off gates and sprues; and a two-wheel electric emery grinder for snagging castings.

A three-ton electric traveling crane is installed for handling the ladles, flasks and castings. The iron storage is underneath the foundry. A large sand pit is provided in the center of the compartment. A thermit welding outfit is carried in the foundry equipment. A sherardizing outfit has also been provided.

The *Optical Shop*, located in the starboard after corner of the superstructure, has the following equipment:

One Brown and Sharpe, No. 1½-A, universal milling machine, driven by a 3-H.P. Reliance motor.

One Pratt and Whitney 10-inch × 5-foot toolmakers' lathe, driven by a ¾-H.P. v. s. Reliance motor.

One Hendey 12-inch geared-head toolmakers' lathe, driven by a 2-H.P. Reliance motor.

One Rivett 8-foot × 22-inch precision lathe, driven by a ¾-H.P. v. s. Reliance motor.

One electric-driven bench-tool grinder.

One electric motor-driven bench buffing lathe.

One Hisey-Wolf electric motor-driven sensitive bench drill.

A small *Drafting Room* has been provided.

The electric-generating and air-compressing plant to furnish the ship with light and power consists of:

Two 85-kw. Turbo-generators, Curtis turbine and Diehl generators.

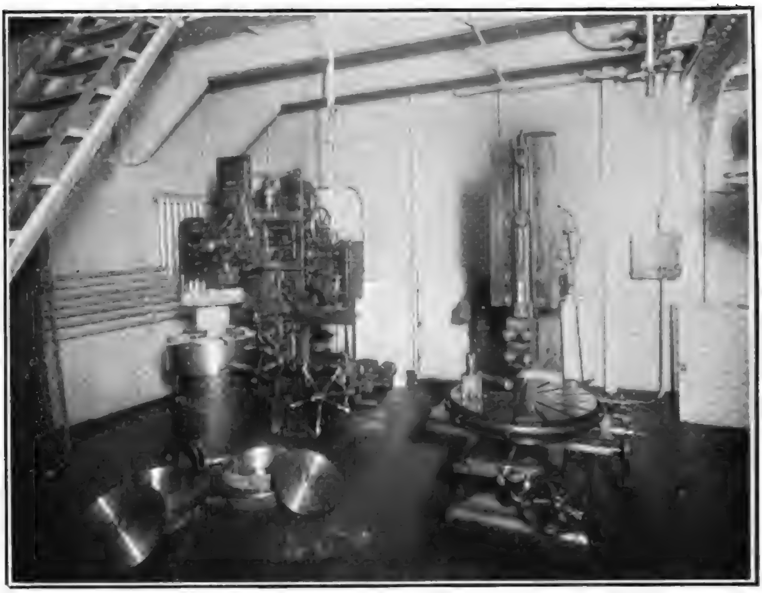
Two 32-kw. General Electric dynamos, driven by piston engines.

One Chicago pneumatic two-stage, compound, steam air compressor, delivering air at 100 pounds pressure throughout the ship. Capacity, 446 cubic feet of free air per minute.

One Norwalk Iron Works steam-driven air compressor for

A Corner of the Upper Platform Deck in the General Store-keeper's Department.

Part of Bar Metal and Pipe Stores.

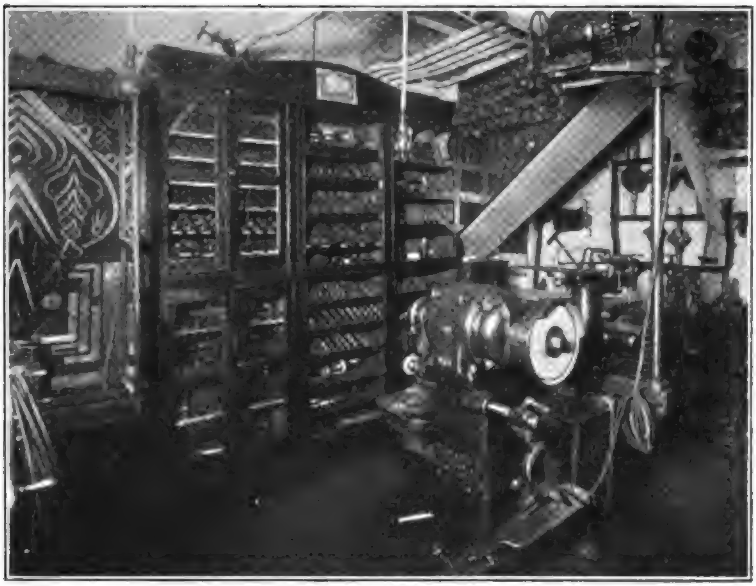


A Corner of the Machine Shop.

Part of the Starboard Gallery of the Machine Shop.

A Part of the Port Gallery of the Machine Shop.

A View of the Center Part of the Lower Deck of the Machine Shop.



A View in the Tool Cage.

After End of Electric Shop.

Coppersmith and Plumbers' Shop.

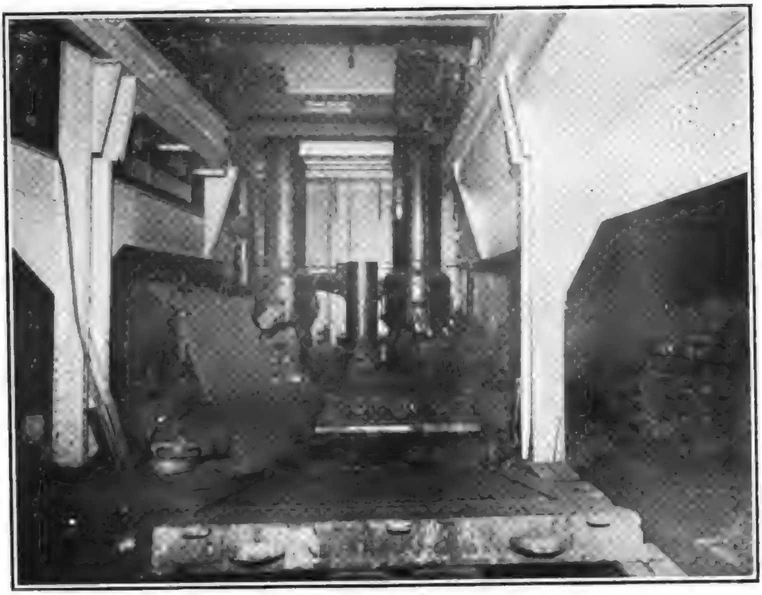
A View in the Forward Pattern Shop.

A View in the Smithy.

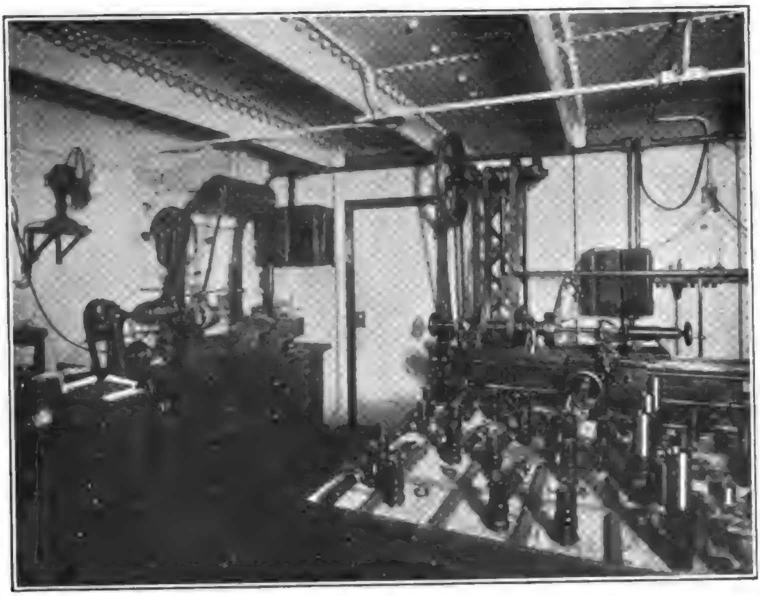
A View in the Smithy.

A Foundry View.

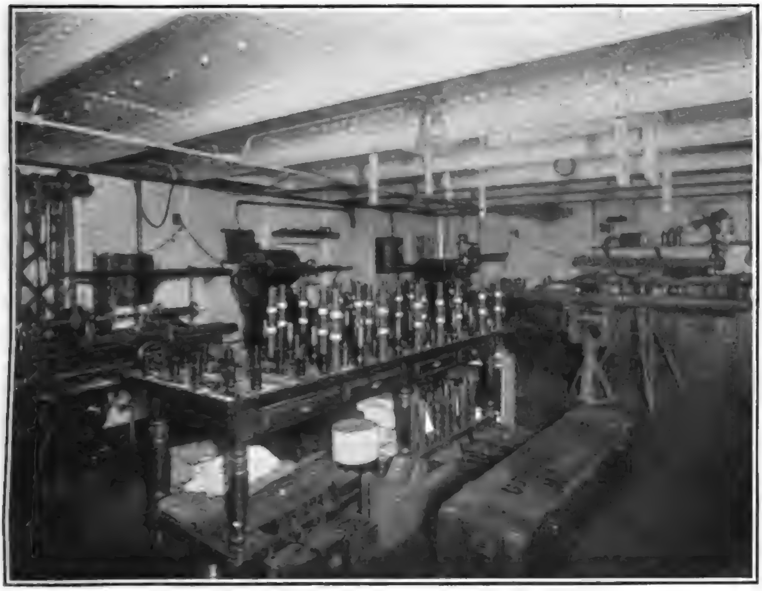
A Foundry View.



A Foundry View.



Optical Shop.



A View in the Optical Repair Shop.

torpedo charging, at 2,500 pounds air pressure, is provided for testing torpedo repairs. Capacity, 20 cubic feet per minute.

The *Vestal* is intended to give prompt repairs to vessels away from the repair yards, the repair ship anchoring wherever the Fleet is temporarily based. Articles to be repaired are usually brought alongside in the ships' boats; when necessary, the vessels themselves coming alongside the *Vestal.* During the five months' stay recently in Vera Cruz, Mexico, eight vessels were taken alongside for more or less extensive repair work.

The repair complement at the present time consists of:

24 Machinists' mates; this includes the optical repair men and the draftsman. The draftsman is also the Progress man.
3 Blacksmiths; this includes the oxy-acetylene operator.
3 Boilermakers; this includes the electric-welding operator.
1 Shipfitter.
4 Moulders.
2 Patternmakers.
1 Carpenter's mate.
6 Electricians.
4 Coppersmiths.
1 Yeoman (clerk).
1 Cook.
20 Helpers.
Total, 70 men.

This force has been found inadequate when with more than a dozen ships. The smaller ships frequently require more repairs than the larger ships, as the equipment of the smaller ships for repair work is usually very limited, and their opportunities for visiting the navy yards are less frequent than the large ships, which have regular overhaul and docking periods.

During seven and one-half months with the Fleet the following principal jobs have been completed:

In the *Optical Shop:*

269 Telescopic gun-sights.
18 Telescopic bore-sights.
25 Spotting glasses and periscopes.
44 Range finders.
78 Telescopes for O.O.D. and Quartermasters' use.
76 Binoculars for Ordnance and Navigation use.
32 Five-inch gun-dotter mirrors resilvered.
81 Stop watches, Ordnance.
41 Clocks, range-keeping and ordinary.
15 Stadimeters, Ordnance and Navigation.
4 Sextants.
4 Azimuth circles.
10 Typewriters.

Small parts for torpedoes, field-gun gear, battery commanders' scales and numerous incidental jobs.

The *Pattern Shop:* Manufactured 229 patterns with necessary core boxes. Much carpenter work being done for the ship, also the sawing and dressing of lumber for various ships of the Fleet. During the ship's stay of five months in Vera Cruz harbor, twelve carpenters' mates and patternmakers were busily employed in the pattern shop, and it was found necessary to extend this shop into the port gallery of the foundry.

The *Foundry:* Made 1,093 iron castings, 1,599 brass and composition castings, 25 monel-metal castings, and 9 aluminum castings. The largest iron castings made weighed about one ton. A six-crucible steel-melting furnace is about to be installed by the foundry force, the necessary drawings, castings, etc., having been made. To date the foundry force has been kept too busy to permit the installation. Two thermit welds on broken iron castings were made. One heat on the sherardizing apparatus was run to demonstrate the possibilities of this system.

The *Copper Shop:* Completed 170 jobs; this included the renewing of the inner shells of three 60-gallon steam-heated galley coppers, the manufacture of the exterior parts (sheet

steel) of sixteen field-artillery ammunition boxes, and about thirty-six ammunition saddle packs for the expeditionary forces in Vera Cruz. About six pairs of main-engine brasses and numerous auxiliary engine brasses were rebabbitted. The largest piece of copper pipe made was a 17-inch bend, the usual jobs being on a 1½ to 10-inch piping. In all cases where the pipes under repairs warranted a renewal, new pipe was made; in other cases, patches were brazed. The copper shop handles all classes of pipe and light sheet-metal jobs received aboard. During the ship's stay in Vera Cruz much overtime work was necessary in this shop, as the majority of the pipe jobs meant that some part of a ship's piping was out of commission until the pipe was returned.

The *Smithy:* This shop, being a combination of blacksmith, shipfitters' and boilermaker shop, handled a total of 534 jobs in the time mentioned; the principal forging jobs being the manufacture of heavy strongbacks, etc., for feed-water heaters, the forging of cargo-boom fittings, pistons for auxiliary engines, the making and repairing of firing tools, etc.

The principal boilermaker jobs were the building of a feed and filter tank for the *Patuxent;* the manufacture of two fresh-water tanks for the *Georgia,* and the renewal of the upper sheets of four of the steam drums of the *Nashville's* main boilers. One hundred and fifty-one oxy-acetylene jobs, and thirty-nine electric welds, were made by this shop. Several of these consisted of repairing ash-ejector piping where the cast-iron pipe was worn through at the bends by the scouring action of the ashes. In these cases a patch of steel plate was cut and bent to conform to the shape of the worn pipe, then the oxy-acetylene operator welded the patch in place. Some B. & W. boiler headers that were found to have developed open seams or pitted hand-hole seats were welded or built up. The jobs of this nature, being always urgent, were rushed through; on one occasion an ash-ejector pipe being patched and returned within one hour of its receipt. The battleship required this prompt action, as she was about to fill her bunkers, the ash ejector pipe passing through a bunker. On an-

other occasion a ship having only one refrigerating engine broke a cross-head. This was gas-welded, and the ship's supply of meats saved.

The *Electric Shop:* Rewound 29 armatures; of these, two 85-kw. and one 200-kw. armatures were the largest ones handled. A number of signaling lights were made; several field coils were rewound. The French cruiser *Des Cartes* availed herself of the *Vestal's* facilities to rewind the field coils of a dynamo. A small electro-plating generator has been provided.

The *Machine Shop:* Carried to completion 624 jobs. Some of these jobs required many machine operations on each article. On one job in mind 40 articles were made, each article requiring 30 machine operations. The variety of work in the machine shop ranged from repairs to artillery-horse collars and motor trucks for the expeditionary forces in Mexico, to work on the torpedoes and main engines of a battleship. The greater part of all jobs handled in any other shop, at some stage of their completion being worked on, in part, by the machine shop.

This being the first year of the ship's commission, a great deal of work has been necessary in making shop equipment. During the five months in Vera Cruz the *Vestal* operated the floating dry dock at that port, docking several of our smaller vessels, renewing bottom zincs, straightening propellers, and giving these ships in general an "overhaul period." In the case of the tug *Patuxent* (755 tons), she was docked, propellers straightened, H.P. cylinders of main engines rebored, new rings fitted, new main air-pump plungers were made to fit rebored cylinders, pumps overhauled, boiler furnace doors made, and so forth. The result was that her speed was increased two knots over what she was capable of making before overhaul. A number of small vessels, British and Mexican, were docked as a matter of courtesy and convenience to the vessels' owners.

A very simple system of job orders is used, giving the routing of the work, appropriation chargeable for material used,

MFB
No.348-14.

U. S. S. GEORGIA,
Veracruz, Mexico,
June 19th, 1914.

From: Captain.
To : Commanding Officer, U. S. S. VESTAL.

SUBJECT: Request repairs by Repair Ship.

REFERENCE: (a) Article 246, Fleet Regulations, 1913.

ENCLOSURES: (a) Blueprint, Georgia's No. 0416.

1. The following work under cognizance of the Bureau of Steam Engineering is requested. This work is beyond the capacity of the ship's force with the regular plant at hand. It is requested that it be completed as soon as practicable.

SPECIFICATIONS.

1. Cast and machine new expander crosshead for 3-ton Ice Machine, as per accompanying blueprint.

Signed. R. E. Coontz.

Original and 1st copy to VESTAL.

1st endorsement. U. S. S. VESTAL, June 23, 1914. Vera Cruz, Mexico.,

To: Commanding Officer, U.S.S. GEORGIA.

1. Copy of job order No.617 is herewith enclosed.

Signed E. L. BEACH.

Exhibit "A."

U. S. S. VESTAL FILE No. 617

JOB ORDER 617

FOR S.E. Dep't. U.S.S. GEORGIA.

CORRESPONDENCE C.O.Let. MBF 4348-14. June 19, 1914.

JOB ORDER ISSUED June 23, 1914.

AUTHORITY /s/ E.L. Beach.
Commander, U.S.Navy,
Commanding.

SPECIFICATIONS

Manufacture 1 expander cross head for 3 ton ice machine.

PLAN OF WORK

Shop P. Make pattern from blue print.

Shop F. Cast from composition.
Shop M. Machine as per blue print.

Copies to:
C.O. GEORGIA.
G.S.K.
File.

Briefs to:
Shop P.

Shop F.
Shop M.

Exhibit B

U.S.S. VESTAL, June 3, 1914.

AUX.# brief.on J.O.# 617

For S.E. U.S.S. GEORGIA.

SPECIFICATIONS.

Manufacture 1 expander cross head for 3 ton ice machine.

Plan.

Shop P. Make pattern from blue print.
Shop F. Cast from compo.
Shop M. Machine as per blueprint.

Exhibit "C"

Estimated date comp'n.
Shop
"
"

Date Com'c'd. 6-26-14.
Date Comp'd. 7-3-14.

Material used in shop P has been charged against this job.

/s/ E.P.Schilling, U.S.N.
Officer in charge.

NOTES.

NAME	RATE	TOTAL HRS.
Bacon	CCM	22

40 ft. 4" white pine pattern lumber.
8 ft. 2" white pine pattern lumber.

Exhibit "C2."

Estimated date comp'n.
Shop
"
"

Date Com'c'd. 7-14-14.
Date Comp'd. 7-23-14.

Material used in shop M has been
charged against this Job.

/s/ J.J.Cotter U.S.N.

Officer in charge.

NOTES.

NAME	RATE	TOTAL HOURS.
Luenzer	MM1c	60

Exhibit C3.

U. S. S. GEORGIA

DATE 7/24/14

JOB ORDER No. 617

RECEIVED THE FOLLOWING ARTICLES:-

Cross head for 3 ton ice machine.

/s/ B. Anderson,
C.M.M., U. S. NAVY.

Exhibit D.

Vestal Form No. 2

U. S. S. VESTAL **N. S. A. TITLE "X" STORES**

Issued 6/26/ , 1914 To Allotment, Ves' Qr.

or to shop for J. O. No. 617 U. S. S. GEORGIA

STOCK NO ENTERED BY STOREMAN	UNIT	NO.	ARTICLES (One item only on each line, Please give full description)	UNIT PRICE
		40	Ft. B.M. 4" pine.	
		8	Ft. B.M. 2" pine.	
			For 1 expander cross /s/ E.P. Schilling.	
			Exhibit E.	

Storeman's Initials | Received | Head of Department | Initials of person receiving stores

Vestal Form No. 2

U. S. S. VESTAL **N. S. A. TITLE "X" STORES**

Issued 6/26/ , 1914 To Allotment, Vest Qr.

or to shop for J. O. No. 617 U. S. S. GEORGIA.

STOCK NO ENTERED BY STOREMAN	UNIT	NO.	ARTICLES (One item only on each line, Please give full description)	UNIT PRICE
	Lb	110	Compo. G.	
		For	1 expander cross	
			/s/ E.P.Schilling.	
			Exhibit E.	

Storeman's Initials | Received | Head of Department | Initials of person receiving stores

Vestal Form No. 5

FOUNDRY REPORT OF HEAT MAD July 1, 191 4 Serial No. 90

MATERIALS USED IN HEAT				PRODUCT OF HEAT.						
METALS	WEIGHT	UNIT PRICE	VALUE	J.O. NO.	NUMBER OF CASTINGS	WEIGHT OF CASTINGS	UNIT PRICE	VALUE	FOUNDRY LOSS	TOTAL VALUE
PIG copper	73	.13429	980	#616 CONNECTICUT	22	70	.12847	899	88	981
PIG tin	32	.3875	1240	#564 NEW YORK	1	50	"	642	65	706
PIG zinc	7	.0634	44	#555 SALEM	4	60	"	771	76	847
PIG										
Scrap copper pipe	100	.1206	1206	#627 FLORIDA	5	45	"	578	57	635
				#394 NEW YORK	1	80	"	1028	100	1128
SCRAP	154	.08	1232	SCRAP (CHARGE TO CLASS 66)		50	.08	400		400
				FOUNDRY LOSS		11				
TOTAL	366		4702	TOTAL		366		4318	384	4702.

CHARCOAL
COKE
SAND
FUEL OIL
TOTAL

Exhibit F.

WEIGHT 366
VALUE 47.02
AVERAGE OR CHARGING PRICE PER LB. .12847

GOOD CASTINGS 305
VALUE OF FOUNDRY LOSS 1.41
DIFFERENCE IN PRICE BETWEEN METAL & SCRAP 2.43
TOTAL 3.84
COST PER LB. .0126

SUBMITTED E. A. Schilling
IN CHARGE OF FOUNDRY

ENTRIES & CHARGES MADE BY [illegible]

date set for completion, etc. A sample of a work request from a ship; a sample copy of the job order issued; a sample of the job order "brief" issued to the shops; a copy of the receipt obtained on delivery; also, a copy of the material stubs used, are hereto attached. Exhibits "A," "B;" "C[1]" (front view), "C[2]" and "C[3]" (back of "C[1]" as rendered by different shops); "D;" "E[1]," "E[2]," "E[3]" and "F." Finally all papers concerned are filed in a large envelope for reference.

Photographs of parts of the shops are shown in cuts.

THE FOUNDRY USE OF NON-FERROUS SCRAP METALS.

By Lieutenant F. M. Perkins, U. S. N., Member.

The following method of using non-ferrous scrap metals at the Puget Sound Navy Yard is the result of some three years of attention to the subject and has proved to be very satisfactory.

Little of originality is claimed for the method or for the principle upon which it is based. It is, in fact, about the only logical way in which to handle the scrap question and one which soon suggests itself to anyone interested in foundry work and is, undoubtedly, now in general use to a certain extent. The following is, therefore, simply a description of a systematic and orderly method of applying an old idea together with some data illustrative of the results of the method.

The primary object of using scrap metals in the foundry is a reduction in the cost of castings, and this must be obtained without a reduction in the quality. These two considerations are not incompatible; they are, on the contrary, complementary when the scrap is properly used.

Before attempting to outline a method of using scrap it may be well to illustrate by a little simple arithmetic the extent of the saving which may be effected by the proper use of scrap. Take scrap gun bronze for example; its composition will be approximately:

	Per cent.
Copper,	88
Tin,	10
Zinc,	2

Now if this scrap gun bronze is not used in the further manufacture of gun bronze, but, through lack of knowledge of its composition or through belief that good bronze cannot be

made from scrap, it is used in the manufacture of some lower-grade alloy, say cast naval brass, we shall have the gun-bronze mixture substituted for an alloy which is only required to be up to the following standard:

	Per cent.
Copper,	62.00
Tin,	0.05
Zinc,	37.95

The loss due to the substitution of the high-grade scrap for naval (yellow) brass is equal to the actual difference in value between the metals composing the two alloys. Let us figure, for example, what the loss would be if 100 pounds of scrap gun bronze were substituted for 100 pounds of naval brass.

*Value of 100 Pounds of Gun Bronze.**

Copper, 88 pounds at $0.16,	$14.08
Tin, 10 pounds at $0.35,	3.50
Zinc, 2 pounds at $0.06,	.12
Total,	$17.70

Value of 100 Pounds of Naval Brass.

Copper, 62 pounds at $0.16,	$9.92
Tin, .05 pound at $0.35,	.02
Zinc, 37.95 pounds at $0.06,	2.28
Total,	$12.22

The difference in value is $5.48 per hundred pounds, or about 5½ cents per pound; or, in other words, gun bronze is worth about 45 per cent. more than naval brass. It is quite apparent that good gun-bronze scrap should not be used for making naval brass.

The loss due to the excess of copper could be eliminated by the addition of zinc and lead in sufficient amounts to reduce

* Prices of these metals are, of course, subject to fluctuations; these are representative of the average.

the scrap gun bronze to the approximate composition of naval brass. The tin, however, which is by far the most valuable component, would cause a considerable loss, as the naval-brass mixture requires practically no tin. Another great objection to this method is that if both scrap gun bronze and scrap yellow brass were to be used in the production of yellow-brass castings the supply of scrap for this purpose would exceed the demand. Furthermore, it is not good practice nor is it economical to take high-grade gun-bronze scrap and convert it to yellow brass by the addition of zinc, and this is due to the fact that the gun bronze is much more free from impurities than yellow brass. Its use in this manner is, therefore, wasteful. It is like putting a four-dollar man on a two-dollar job.

Gun bronze and naval brass have been taken as an illustration; the foregoing remarks apply also to several other high copper-bearing alloys related to gun bronze and several comparatively low copper-bearing alloys related to naval brass.

The method in most common use and the one recommended is, briefly, to segregate the scrap according to composition, to melt it in large quantities, to obtain homogeneous lots, to pig it, analyze it and to use it, with the proper additions of virgin metals, in the manufacture of the alloy which it most nearly approximates. This method, properly applied, has no apparent ill effects upon the alloys, but, on the contrary, seems to improve their physical properties and casting qualities.

The first and a most essential point is the careful segregation of the scrap according to composition. The scrap is delivered to bins built alongside the yard railroad track within a few yards of the foundry. Separate bins are provided for the stowage of scrap copper, zinc, lead, gun bronze, phosphor bronze, manganese bronze, yellow brass, and separate bins for the borings and turnings of each kind. Metallic-packing scrap is stowed in the foundry and babbitt scrap is kept in the machine shop on the babbitting floor. Non-ferrous scrap coming off ships under repair is delivered in cars to the bins. Here the master molder, or a leading man, looks over the scrap

and supervises its distribution into the various bins. It is a comparatively easy matter for an experienced and intelligent molder to properly separate the scrap. From the nature of the casting he can tell of what composition it was originally made and can also readily determine by the color whether it is yellow brass or "red brass." The latter term applies to high copper-bearing alloys such as gun, valve and phosphor bronze. No trouble has been experienced in properly classifying scrap of this nature. Scrap castings containing pieces of iron or steel, such as bolts, nuts, studs, etc., are thrown to one side and the iron or steel removed before placing the scrap in the bins. Copper pipe is first taken to the blacksmith shop, placed under a hammer and flattened and then cut in the shears into convenient lengths for charging. Heavy copper-wire condenser tubes, long pieces of sheet copper, sheet brass, etc., are also cut in the shears to convenient lengths. Light sheet copper and light copper wire are heated in an open wood fire and tamped into shape for charging. Heavy castings which are too large for charging are heated and broken up with a sledge or the drop.

The proper segregation of scrap borings and turnings must commence in the machine shop where they accumulate. A removable pan for receiving borings and turnings is placed under each machine. The leadingmen are held responsible for seeing that the pan is changed or emptied when the kind of material to be worked on in the machine is changed. Several stations are provided for receiving the borings and turnings, each station having several labelled boxes capable of holding several days' accumulation of the various kinds of turnings. The turnings are collected about twice a week, run through a magnetic separator to rid them of what iron and steel have found their way in, and then delivered in the boxes to the foundry bins.

In handling borings and turnings special care is taken to guard against the introduction of iron or steel filings, drillings, turnings, etc. Sweepers are not allowed to use the scrap boxes for receptacles for their sweepings, as these always contain

iron and steel filings and dust. Too much must not be left to the magnetic separator; it removes a large proportion of the iron and steel, but not all. The borings and turnings from babbitt and metallic packing also require care in handling; these metals are expensive and in this form are easily spoiled. Little chips of iron and brass will not melt down when these white metals are remelted, and they form hard spots which will score a rod or journal. Brass borings in particular must be kept out of white metals for, of course, they cannot be removed by the magnetic separator. Metallic packing and babbitt must not be allowed to become mixed; the two metals are difficult to distinguish by sight and are usually of widely differing composition.

A little care in the machine shop will pay big dividends in the foundry in the form of increased purity and consequent increased value of the scrap. A handful of iron will ruin a hundred pounds of babbitt.

As the various scrap metals accumulate in sufficient quantities in the foundry bins they are pigged; the cost of pigging and the melting loss being charged to title Z orders issued upon request of the general storekeeper. These charges are prorated on the metal pigged on a per pound basis. Scrap metals are pigged in large quantities (about 25,000 pounds and over) at a cost of about ½ cent per pound plus the melting loss. This loss varies considerably with different grades of scrap, and will run from two or three per cent. in alloys containing little or no zinc to eight or nine per cent. in alloys high in zinc.

In figuring the actual money loss due to volatilization, furnace loss or "shrinkage," as it is variously called, it should be remembered that, in the case of metals high in zinc, the metal which is lost is not of the same composition as the alloy being melted, but contains relatively a much higher amount of zinc. As zinc is the cheapest metal found to any extent in the brasses this reduces the actual money loss to considerably less than the percentage of loss by weight would seem to indicate. For example: Suppose that in melting 100 pounds of yellow

brass containing copper 60 per cent., zinc 40 per cent., valued at $12.00, or 12 cents per pound, the melting loss amounts to 8 per cent., or 8 pounds. The actual loss then is not 8 per cent. of $12.00, or $0.96, but approximately 6 pounds of zinc at $0.06 per pound plus 2 pounds of copper at $0.16 per pound, or $0.68. Thus, while the melting loss by weight amounts to 8 per cent. the actual loss in money is only 5.7 per cent. This difference is not at present considered in accounting. In the case of melting scrap for pigging the loss by weight is never actually as great as the weights of a given amount of metal before and after pigging would seem to indicate. Nearly all scrap contains a certain amount of non-metallic foreign material of no value, in the form of dirt, oil, grease, sand, insulating materials, etc. In certain kinds of scrap this may form quite an appreciable amount, and the cheapest and most practicable way to remove it is by fluxing it out in melting.

The primary object of pigging scrap is to reduce it to lots of homogeneous and known composition so that it can be intelligently used in the manufacture of castings. That the actual saving in dollars and cents far outweighs the cost of pigging will hardly be disputed by anyone who has had practical experience with the foundry-scrap question. Beside the direct saving there are other practical benefits resulting from pigging and melting, by analysis, namely:

1. The metal will be used for the purpose for which it is best suited.

2. A more uniform grade of castings can be produced.

3. It affords an opportunity to cast test pieces for determining the physical properties and casting qualities of the metal before it is made into castings.

4. The scrap can be fluxed and cleaned before it is used for castings.

5. Great reduction in stowage space.

6. If records are kept they provide a means for determining the causes of defects and afford valuable data for continuous improvement.

7. It tends toward increased order and system and more scientific melting in the foundry.

There are a few cases wherein the scrap, by its nature, affords most of the information which is ordinarily obtained by pigging, and in such cases there is no need to take the melting loss due to pigging. If a certain large amount of scrap is known to be of homogeneous composition there is no need to pig it. Condenser tubes, for example, form a continuous source of scrap of unvarying composition, and for this reason should be stowed separately. A large bronze propeller or any other large, single casting can be broken up, a sample taken and analyzed, and the metal used to the best advantage without being pigged. Such cases, however, form but a small percentage of the total amount of scrap.

Copper pipe is a very valuable and highly prized form of copper scrap and is used without pigging. It is made of a high grade of copper (99.5 per cent. pure) and, either alone or mixed with virgin ingot, produces excellent brass and bronze castings. Scrap lead, if clean, is used directly; if dirty it is melted and cleaned. The two principal sources of scrap zinc are old hull zincs and the trimmings, cuttings and drillings from new zinc. The former contain a very large amount of dross while the latter are very pure. The old zinc is run down and the dross skimmed off; the new zinc is used directly.

The scrap, having been properly separated and prepared for charging, is melted in a two-ton, oil-fired Hawley-Schwartz furnace in lots of from 3,000 to 5,000 pounds, the capacity of the furnace depending largely upon the shape and density of the scrap. As much care is exercised in the melting of the scrap as in the melting of metal to be used directly in the molds, and this point is of great importance. A charge of a few hundred pounds of scrap and two or three pounds of charcoal is first placed in the pre-heated furnace and the heat turned on. When this charge begins to melt down the molten metal is covered with a heavy layer of charcoal and the remainder of the scrap is charged. The molten metal in the

bottom of the furnace, with its covering of charcoal, forms a bath into which the metal runs as it melts down.

The function of the charcoal is to prevent oxidation of the metal. Oxidation or burning of the metal, in addition to causing a high furnace loss, produces blow-holes, due to absorption of the oxide in the metal and flaws due to the mechanical inclusion of the hardened metallic oxide.

When molten metal is exposed to the air a film of oxide forms upon the exposed surface, the depth of the film and the rapidity of formation depending somewhat upon the temperature, but primarily upon the affinity of the metal for oxygen. This film, when cooled slightly, becomes a hard scale, and if allowed to enter the mold produces flaws in the casting.

Of all the metals used in the brass foundry copper has the greatest affinity for oxygen and it is also the most widely used; hence the necessity for a working knowledge of oxidation and its prevention.

Charcoal reduces oxidation by forming a covering over the molten metal and thereby excluding air. It also reduces the absorption by the metal of the sulphur in the fuel. All the fuels used in brass melting (coal, coke and oil) contain sulphur, which should not be allowed to enter the brass. Charcoal contains no sulphur, but is too expensive to be used as a fuel except in unusual cases.

In order that the charcoal may properly serve its purpose it should be broken up in small pieces; the smaller the pieces that can be used in a furnace or crucible without being blown out by the blast the greater will be the protection to the metal and the less the expenditure of charcoal.

A flux of borax, rock salt or glass is used to fuse with the dirt and impurities in the scrap and form a slag on the surface of the metal, which is later skimmed off. The amount of flux to be used depends upon the cleanliness of the scrap and upon the amount of zinc contained in the metal, high zinc metals requiring more flux.

The molten scrap is poured into a 1,000-pound ladle swung from the crane, and from this ladle is poured into a reversible

FIG. 1.—REVERSIBLE MOLD FOR CASTING SCRAP BRASS AT FOUNDRY.

FIG. 2.—REMELTED SCRAP BRASS AT FOUNDRY (ABOUT 110 TONS).

ingot mold having a capacity of twenty ingots of about forty pounds each. Fig. 1 shows the ingot mold, and in the background of the same figure is the furnace used for melting the scrap. After the furnace is well heated 4,000 pounds of metal can be melted in 45 minutes and cast into ingots in about 30 minutes.

The metal is thoroughly stirred in the furnace and again stirred and fluxed with borax in the ladle and skimmed before being poured into the ingot molds. Thorough stirring, both in the furnace and ladle, is of the greatest importance, as it insures uniformity of composition of the entire heat. The serial number of the heat is then stamped on each ingot for purposes of identification and to prevent the possibility of lots becoming mixed. The various lots are then stowed separately in bins with the serial number of each lot posted on the bin containing it. Figure 2 shows about 100 tons of pigged scrap stamped and ready for stowage in the bins, which are about twenty feet away, but not shown in the picture. The picture gives a good idea of the great saving in stowage space due to pigging.

Standard test pieces are cast, and these and a mixture of drillings from three ingots are sent to the chemist for determination of the composition, tensile strength and elongation. The chemist forwards his results to the shop superintendent, who determines from the data what alloy the scrap is best suited for and computes the amount of new material to be added to the scrap to bring it up to the desired composition.

These results are entered on the "Scrap-Metal Record" card forwarded to the master molder, who files the cards for reference when using the scrap.

Fig. 3 shows this card with entries for a heat of scrap valves and pressure castings. It will be noticed that the card also shows the analysis, tensile strength, elongation and melting loss. The scrap is used in the foundry in accordance with the instructions on the card, and in one or more heats for each lot of scrap used test bars are taken, and the tensile strength, elongation and chemical analysis of the metal after

recasting are entered on the back of the card. The comparison of the properties of the metal before and after being "sweetened up" with virgin metal affords interesting and valuable information. A little experience will enable one to accurately control the composition of the brass-foundry products and to predict very closely what the properties of a given lot of scrap will be when corrected and made into castings. The card gives a fairly complete record for practical foundry purposes and enables the foundryman to do his melting scientifically, accurately and without guess-work.

SCRAP METAL RECORD

HEAT NO	DATE	KIND OF SCRAP MELTED
2	11/19/14	Comp. G - Heavy valves and pressure castings.

CHARGED	RECOVERED	LBS LOSS	PER CENT LOSS
4800	4695	105	2.2

ANALYSIS

Copper	86.64	Antimony		TENSILE STRNGTH	41750
Tin	8.87	Manganese			
Lead	0.64	Phosphorus		ELONGATION	6.25
Zinc	4.05	Iron	Trace		

This Scrap is best suited for Comp. G - Gun Bronze

To obtain same, add the following to each 100 lbs. of ingot:

COPPER	TIN	PHOS. TIN	LEAD	ZINC	ANTIMONY		
78	8½	1½	—	—	—		

FIG. 3.

A careful study and conscientious use of such records and a little study of the principles of furnace and crucible operation will soon permit one to use a considerable percentage of scrap and to improve the quality of the metal at the same time. All that is required of the melter are average intelligence and absolute obedience in following instructions. Many melters. like to surround their operations with mystery, and some of them have their own brand of jealousy-guarded "dope" which they introduce into the ladle and whose principal effect is to

create a large cloud of smoke and many admiring glances from the helpers and apprentice boys. While experience in a melter is desirable it is not essential. Under proper supervision any intelligent, willing man interested in learning can be made into a first-class brass melter in three months.

The principal points to be cared for in good practice are care in the preparation of the scrap, adherence to the specifications governing the composition of the metal, and the observance of a few well-known principles of melting. The most essential of these principles are:

1. Protection of the metal from oxidation during melting. This is ordinarily accomplished mechanically by covering the bath with charcoal or by otherwise preventing the flame and air from coming in direct contact with the metal, as by the use of covered crucibles.

2. Deoxidation of the metal after melting. Oxidation cannot be entirely prevented during melting. After melting it can be deoxidized chemically to a certain extent by the addition of some substance having a greater affinity for oxygen than the alloy. One of the best agents for this purpose is from .75 per cent. to 1 per cent. of phosphor tin (a mixture containing 95 per cent. tin, 5 per cent. phosphorus). Phosphor copper is used when the metal to be deoxidized does not contain tin. As phosphor copper contains 10 per cent. phosphorus only half as much need be used. It is a mistake to assume that the more phosphorus is introduced the cleaner the metal will be. If the proper amount is used it will take up the oxygen and pass off as gas and the metal will show barely a trace of phosphorus or none at all. If too much is used only that which combines with the oxygen will pass off and the remainder will remain in the metal and cause blowholes.

3. Melting the metals in the inverse order of their fusibility and volatility, *i. e.*, melting that metal first which is the least volatile and has the highest melting point. There are a few metals which are more volatile than others having a lower melting point and, in order to reduce the melting loss, the

volatility should, in such cases, be the first and fusibility the second consideration. Zinc is the most common example of this met with in foundry practice; it has a higher melting point than tin or lead but is much more volatile than either and is added last.

4. Thorough mixture of the metal composing the alloy. This is accomplished mechanically by means of a stirring rod used in the furnace and also in the ladle. As each new metal is added it should be thoroughly stirred as soon as it has run down. The last operation before pouring the mold should be to stir the metal and, in case of the white-metal alloys, it will be found beneficial to continue gently stirring the metal in the ladle during pouring. A metal which has been found to contain pin-holes, blow-holes or flaws can often be made perfectly good simply by remelting.

5. Pouring the metal at the proper temperature. This is important but one of the most difficult points to control. The proper temperatures of pouring must first be known and some quick and reliable method of measuring them must be provided. Correct casting temperatures have been determined for a few metals but not for the majority of them. A scientific investigation of the melting temperature and casting temperature upon the principal alloys would be of the greatest value. The main points to be covered would be the effects upon strength, elasticity, hardness and soundness. The alloys investigated should not be chemically pure, but should contain the impurities to the extent ordinarily found in actual practice. The metal should not be overheated and allowed to cool; this practice increases the furnace loss and produces inferior castings. The metal should be removed from the furnace when the proper temperature is reached and poured into the molds as soon as possible. In general, the brasses and bronzes should be poured hot and cooled quickly; rapid cooling or quenching has the opposite effect upon brasses to that produced by the same means upon iron. Rapid cooling or chilling a brass or bronze casting increases its strength and decreases its hardness. Various makes of electrical and op-

tical pyrometers are on the market for determining the temperatures of molten brass. A great many of them are inaccurate or unpractical. Investigation to determine the best pyrometer for brass-foundry use would be of value after proper casting temperatures had been determined.

6. Removal of dirt, dross and foreign matter by the use of suitable fluxes. The fluxing material—borax, salt or salammoniac—is thrown upon the surface of the metal in the ladle and well stirred in. The resulting dross or slag which forms on the surface is then skimmed off.

It has been found from experience that a very high percentage of scrap can be used if it is clean and well suited to the alloy in which it is used. In fact, high-grade scrap alone with the addition of sufficient zinc to replace the loss due to volatilization will often make excellent castings entirely complying with the chemical and physical requirements of the specifications. To take proper advantage of such high-grade scrap it is necessary that the method of pigging and melting by analysis be followed. In the case of the high copper-bearing alloys (gun bronze, phosphor bronze, valve bronze, etc.), the greatest difficulty encountered in the use of scrap is to keep the iron at the very low point required by the specifications, hence the necessity, previously mentioned, of taking every precaution to remove iron from scrap castings and to prevent it being mixed with borings and turnings. Lead also gives trouble and will be referred to later.

In order to reduce the relative amount of iron it is often necessary to add considerably more new metal to a given amount of scrap than would be required if the scrap contained less iron. Experience would seem to indicate that the allowable percentage of iron could be increased without detriment to the metal. The presence of a small amount of iron in gun, valve and phosphor bronze within certain limits seems to cause a slight increase in tensile strength and a slight decrease in elongation. As the amount of iron is increased the tensile strength and elongation decrease. This statement is not based upon accurate, scientific experiment, but upon data collected

in the ordinary course of foundry operation. It would be interesting, instructive and of actual value to determine by exact laboratory methods the actual effect of varying percentages of iron upon several of the bronzes with the object of increasing the allowable maximum if found advisable.

The following table shows the composition, tensile strength and elongation of five heats of scrap containing iron:

Heat No.	Cu.	Sn.	Zn.	Pb.	Fe.	T. S.	Per cent. elong.
1	87.02	8.65	3.94	0.15	0.24	39,000	8.6
2	86.64	7.90	4.56	0.69	0.21	42,550	10.5
3	85.92	7.90	5.30	0.75	0.13	39,475	7.8
4	85.60	6.78	5.97	1.65	Trace.	27,730	14.0
5	71.88	1.57	25.93	0.28	0.34	31,100	18.8

Numbers 1, 2 and 3 are suitable for conversion to gun, valve or phosphor bronze, but, on account of the comparatively large amounts of iron in the scrap, it will be necessary to add an excessive amount of new metal to reduce the percentage of iron to the point required by the specifications. Therefore, the scrap will form but a small percentage of the total melt and the saving due to its use will be comparatively small. Number 4 contains only a trace of iron but is rather high in lead and zinc. The addition of 59 pounds of copper, 7 pounds of tin and 1 of phosphor tin as a deoxidizer to the number 4 scrap will convert it into good valve bronze, the above amounts being added to 100 pounds of the scrap. In this case the resulting metal will contain 60 per cent. of scrap, and the saving due to its use will be considerable. This illustrates the necessity of keeping iron out of brass scrap and shows the saving that can be effected thereby. Number 5 is too high in zinc to be suitable for anything but yellow brass and should be converted to such by the addition of proper amounts of lead and zinc.

The following table shows the composition, tensile strength and elongation of thirteen heats of gun and valve bronze which contain, on the average, about 50 per cent. of new metal and 50 per cent. of scrap.

Heat No.	Cu.	Sn.	Zn.	Pb.	T. S.	Per cent. elong.
1	86.64	9.38	3.03	0.95	35,240	24.2
2	87.51	8.75	3.64	0.10	38,610	25.0
3	87.26	8.33	4.01	0.40	34,240	19.5
4	86.72	8.80	3.59	0.89	39,400	18.75
5	88.42	8.76	2.32	0.50	38,700	21.9
6	88.36	7.67	3.24	0.48	36,260	25.0
7	87.28	7.28	4.01	1.43	34,370	18.75
8	89.03	7.67	2.49	0.81	36,480	25.0
9	87.85	7.56	4.09	0.50	32,630	15.6
10	88.59	7.61	3.21	0.59	35,400	20.3
11	88.39	7.45	3.55	0.61	34,370	18.75
12	88.26	7.37	3.75	0.62	34,870	18.75
13	88.11	8.02	2.98	0.89	35,240	20.3

In only one heat are the specifications complied with (No. 2) for gun bronze in so far as the lead content is concerned (not over 0.20 per cent.). The others meet the specifications for valve bronze. The tensile strength and elongation are, however, uniformly excellent and in every case exceed the requirements of gun bronze.

The following table shows the results obtained from eight heats of gun bronze made entirely from first-grade ingot (new) metal:

Heat No.	Cu.	Sn.	Zn.	Pb.	T. S.	Per cent. elong.
1	87.36	10.01	2.54	0.09	30,940	10.1
2	87.44	10.24	2.22	0.10	33,100	11.7
3	87.45	10.17	2.22	0.16	32,770	12.5
4	87.20	10.37	2.32	0.07	30,920	10.9
5	87.20	11.00	1.64	0.16	31,460	8.6
6	86.59	10.87	2.47	0.07	31,380	9.4
7	87.70	10.21	2.06	0.03	34,370	14.1
8	87.20	9.73	3.03	0.04	31,520	12.5

Note that the requirements of the specifications are not met as regards elongation. This is not offered as proof that 15 per cent. or more of elongation cannot be obtained in gun bronze, for by careful melting this percentage can be obtained. Comparison of the two preceding tables does show, however, that for the same melting practice better metal can be made by the proper use of good scrap comparatively free from impurities. The scrap should be of the approximate composi-

tion of the alloy being made and may be used to advantage in amounts varying from 10 per cent. to 60 per cent. It has been found that, when properly used, scrap not only improves the strength and elasticity but the soundness of the castings as well, and soundness is often the primary consideration. The writer has talked over this matter with a number of foundry foremen, and in each case his own experience has been confirmed, *i. e.*, that it is possible to produce better castings when a certain amount of good scrap is used. The explanation of this probably lies in the fact that pure copper absorbs oxygen and certain gaseous products to a much greater extent than an alloy which results in an increased porosity. This a well-known fact to which the great difficulty of obtaining sound copper castings bears evidence.

It is a difficult matter to obtain scrap in very large quantities sufficiently free from lead for use in alloys where the maximum amount allowed is 0.20 per cent. The elimination of iron is largely a matter of care in handling the scrap, but lead is not so easily removed. In the above table it will be noticed that the lead runs as high as 0.16 per cent. when no scrap is used. This is due to its being contained as an impurity in the other metals.

Foundry results appear to indicate that the amount of lead could be increased with benefit to gun bronze and number one phosphor bronze, in that it would permit a wider use of scrap with the beneficial results before noted, at the same time reducing very materially the cost of the metal.

The general effect of small quantities of lead upon the brasses and bronze is to decrease the hardness and increase the density. In other words, machining qualities are improved and porosity is decreased. Lead is used in valves and pressure castings to make them "close," *i. e.*, to decrease the porosity. Too much lead reduces the strength, but the amount of lead that can be added without a material reduction in strength is fairly high. There are commercial valves, for example, made of valve metal containing 3 per cent. lead and

having a tensile strength of 30,000 pounds and an elongation of 20 per cent.

As previously mentioned when considering the effect of iron, the foregoing tables and discussion concerning lead are not based upon scientific experiment, but upon observations made and data collected in the ordinary course of foundry operation.

John Dewrance, in a paper entitled "Bronze," read before the Institute of Metals and later published in "Engineering" and republished in the JOURNAL OF THE A. S. N. E. for May, 1914, furnishes some very interesting information concerning the effects of lead upon gun bronze tested at different temperatures. He found that straight gun bronze (88–10–2) dropped from a tensile strength of about 32,500 at 350 degrees F. to 19,000 pounds at 400 degrees F. Upon substituting ½ per cent. of lead for an equal amount of copper (giving 87½, 10, 2, ½) he found that the strength of the bronze remained at about 32,000 pounds up to a temperature 550 degrees F., above which it rapidly decreased to 22,500 pounds at 600 degrees F. The variation in elongation he found to be even more pronounced; at 550 degrees F. the alloy containing ½ per cent. lead gave an elongation of 18 per cent., and the straight gun bronze at the same temperature gave an elongation of a trifle less than 2 per cent.

In gun bronze to which had been added 10 per cent. of lead at the expense of the copper was found a strength of 28,500 pounds and an elongation of 10 per cent., and when the amount of lead was increased to 16 per cent. the alloy still had a strength of 25,000 pounds and elongation of 5 per cent. This test was made with the metal heated at 500 degrees F.

In considering these results it should be borne in mind that lead softens the brasses, and therefore a casting which is to be subject to great erosive action should contain but very little lead.

The most authoritative writers on the subject of alloys have usually investigated pure alloys and have taken particular care to see that impurities were eliminated. For this reason

reliable information of an exact nature as to the effects of impurities upon brasses and bronzes is very scarce. It is believed that careful investigation of the effects of the use of scrap and of varying amounts of lead and iron in certain alloys used in the Navy would corroborate results obtained in the foundry and disclose information which would warrant a wider use of scrap in the higher-grade alloys.

THE GAUSS GRAPHIC METHOD OF REPRESENTING LENSES AND TELESCOPES.

BY LIEUTENANT F. J. CLEARY, U. S. NAVY, MEMBER.

The exact mathematical calculations required in designing lenses are complicated and extensive, especially in new construction, the optical layout of a new design of telescope covering a dozen or more pages of logarithmic work.

The formulae used and the resulting calculations are so technical in character that their presentation here is not warranted.

There is, however, a graphic method of lens calculation which is so simple and which illustrates so clearly the relations existing in a telescope system between angle of field of the instrument, angle of field of the eye piece, magnification, aperture of the objective lens, exit pupil, focal lengths of objective lens and eye piece, and the relative positions of the various lenses and the images formed, that it is believed that a description of this graphic method will be of interest to the naval service.

This method of lens representation is due to C. F. Gauss, one of the foremost German mathematicians and opticians, and consists of substituting for the curved-lens surfaces a series of planes normal to the central axis.

A few preliminary definitions and explanations are necessary.

A *lens* is a portion of a refracting medium bounded by two surfaces of revolution called *faces*. The face that is presented to the source of light will be called the *first face*. The other face will be called the *second face*. The lenses with which we have to deal are made of glass, and the faces are either both spherical or one face plane and the other spherical.

The *principal axis* of a lens with two spherical faces is the straight line joining the centers of curvature of its faces. In a lens that has one plane face, the principal axis is the normal to the plane surface that passes through the center of curvature of the spherical face.

The points where the principal axis cuts the faces of the lens are called *poles*.

The distance between the two faces measured along the axis is the *thickness* of the lens.

Lenses are divided into two general classes:

First, those that can form real images, called convergent, positive or collective lenses. They are generally thicker in the middle than at the edges.

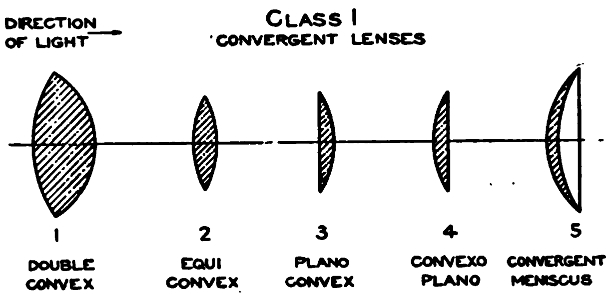


FIG. 1.

Second, those that, when situated in a medium less dense than the lens, can form only virtual images. Such lenses are called divergent, negative or dispersive lenses. They are generally thinner in the middle than at the edges.

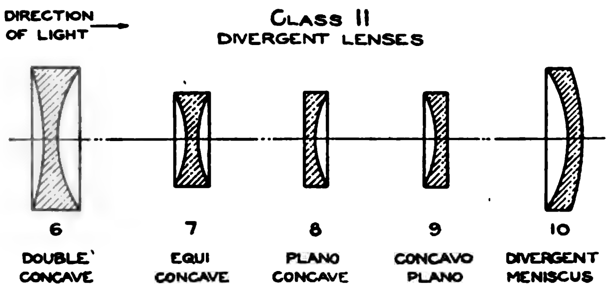


FIG 2.

The *optical center* of the lens is the point of intersection on the axis made by any ray whose direction after emergence is parallel to its direction before entering.

The *first principal point* is the focus on the axis *conjugate* to the optical center by refraction through the first face alone.

The *second principal point* is the focus on the axis conjugate to the optical center by refraction through the second face alone.

The first and second principal points may also be defined as the points on the axis intercepted by the prolongations of the entering and emergent directions of a ray of light passing through the optical center.

The *equivalent thickness* is the distance between the first and the second principal points measured on the axis.

These definitions are illustrated by Figs. 3 and 4.

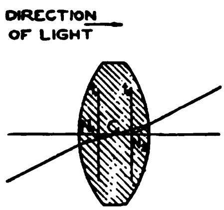


FIG. 3.

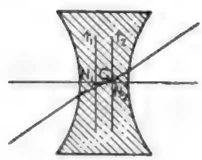


FIG. 4

C is the optical center; N_1 and N_2 are the first and second principal points; p_1 and p_2 are the first and second principal planes; N_1N_2 is the equivalent thickness. Where the faces of the lens are not of the same curvature the principal planes are displaced toward the face having the greatest curvature (least radius of curvature). Where one face is plane, one principal plane always lies tangent to the curved face, the other principal plane lying within the lens. In meniscus lenses one principal plane always lies outside of the lens. If the meniscus is deep enough both planes will lie outside the lens (Fig. 5).

The *first principal focus* (f_1) of the lens is the focus on its axis conjugate to a point at an infinite distance in rear of the lens.

The *second principal focus* (f_2) of a lens is the focus on its axis conjugate to a point at an infinite distance in front of the lens.

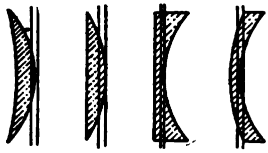


FIG. 5.

The term "conjugate" is used in optics to express the relationship between two mutually interchangeable factors.

For example, in Fig. 6, f_1 and a point at an infinite distance in rear of the lens are conjugate for the reason that rays of light from a source at f_1 will intersect at infinity, and all rays of light from a source at infinity will intersect at f_1. Similarly, in Fig. 16, O and I are conjugate, for the reason that an object at O will form an image at I and an object at I will form an image at O.

The principal foci are illustrated by Figs. 6 and 7.

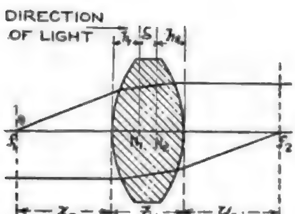


FIG. 6.

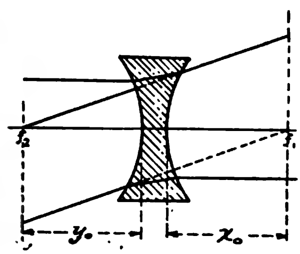


FIG. 7.

f_1 and f_2 are the first and second principal foci. The planes through the foci normal to axis are, respectively, the first and second focal planes.

Where the faces of the lens are of equal curvature the distance between the first principal focus and the first principal point is equal to the distance between the second principal focus and the second principal point. This distance is called the focal length. Where the curvature of the two faces is not the same the focal lengths of the lens are not equal.

The principal use of the principal planes is derived from the property that the prolongation of both the incident and the refracted rays cuts the principal planes at the same distance from the axis. If we know the equivalent thickness and the focal lengths of a lens the lens may be represented by the principal planes only, and a graphic solution may be worked out for a lens or a series of lenses. The proof of this method will be given at the end of this article.

The equivalent thickness (δ) and the focal length(f) can be obtained as follows:

In Fig. 6,

Let

h_1 be the distance of the first principal point (N_1) from the first face;

h_2 be the distance of the second principal point (N_2) from the second face;

t be the thickness of the lens on the axis;

$r_1 r_2$ be the radii of curvature of the first and second faces;

μ be the refractive index of the glass;

Then

$$\delta = t - (h_1 + h_2).$$

The values of h_1 and h_2 for a double convex lens are:

$$h_1 = \frac{t r_1}{\mu (r_1 + r_2) - t (\mu - 1)};$$

$$h_2 = \frac{t r_2}{\mu (r_2 + r_1) - t (\mu - 1)};$$

And

$$\delta = t - \frac{t (r_1 + r_2)}{\mu (r_1 + r_2) - t (\mu - 1)}. \quad . \quad . \quad . \quad (1)$$

If x_0 is the distance from the first principal focus to the first face,

y_0 the distance from the second principal focus to the second face,

then from the definition of focal length given above

$$f = x_0 + h_1 = y_0 + h_2,$$

and we have the formula

$$\frac{1}{f} = p = (\mu - 1)\left(\frac{1}{r_1} + \frac{1}{r_2}\right) - \frac{t(\mu - 1)^2}{\mu r_1 r_2}.$$

For practical purposes the thickness (t) of the lens may be disregarded and the formula becomes

$$\frac{1}{f} = p = (\mu - 1)\left(\frac{1}{r_1} + \frac{1}{r_2}\right).$$

The following numerical example is given:

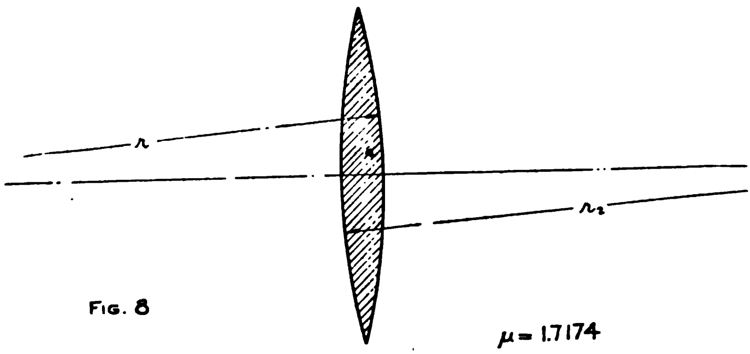


FIG. 8

$\mu = 1.7174$

$r_1 = r_2 = 200$ mm $= .2$ M
CURVATURE $= \frac{1}{.2} = 5$ DIOPTRIES
TOTAL CURVATURE $= 10$ DIOPTRIES

$$\frac{1}{f} = p = .7174 \times \left(\frac{1}{.2} + \frac{1}{.2}\right) = 7.17.$$

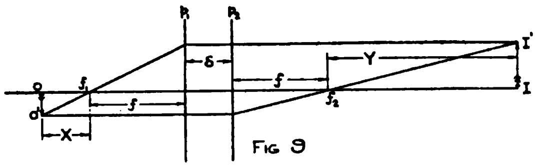
$$f = \frac{1}{7.17} \text{ meters} = 139.4 \text{ millimeters.}$$

The term "p" in the above formulae is known as the *power*. This must not be confused with *magnifying power* or *magnification*, as the terms have entirely different meanings.

The *power* of a lens is its ability to change the curvature of the advancing front of a wave of light.

The *power* of a lens is the reciprocal of its focal length. The international unit of power is the dioptrie. The power of a lens in dioptries is the reciprocal of its focal length in meters.

Fig. 9 shows the graphic solution of a lens (or a lens equivalent to a compound lens) where the focal lengths and the equivalent thickness are known.



p_1 and p_2 are the principal planes, f_1 and f_2 are the first and second principal foci, and OO^1 is the object at a distance X from f_1. If we take the point O^1 on the object and draw a ray parallel to the axis we know from the definition of principal foci that this ray will pass through f_2, the second principal focus. We know also that a ray drawn from O^1 through f_1, the first principal focus, will emerge parallel to the axis, and therefore the intersection of these two rays at I^1 will give a point of the image conjugate to O^1, and an image, real and inverted will be formed at II^1. The distance Y of this image from the second focal point is given by the very simple relation

$$XY = f^2.$$

This equation shows that when $X = \infty$, $Y = o$ and conversely, which we know to be true from the definition of the principal foci. Also if X is large, say one mile, then Y is so small as to be inappreciable. Hence, any pencils of light from a distant object can be regarded as parallel pencils.

An approximate value of the magnification of a lens in the above terms is given by

$$M = \frac{f}{X}.$$

Another property of lenses is that parallel pencils that are not vertically incident, but make a small angle with the axis come to a focus in the focal plane.

COMBINATION OF LENSES.

The effect of combining lenses may be determined by using the principal planes, and the theory, already stated, that parallel pencils that are nearly parallel to the axis are focused on the focal plane. This enables us to trace any ray through a lens as follows:

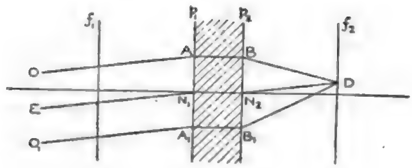


FIG. 10.

p_1 and p_2 are the principal planes, f_1 and f_2 the principal focal planes, and N_1 and N_2 the unit points. Draw OA to represent any ray incident on the first principal plane at A. We know that it emerges at B with AB parallel to the axis. This gives one point B in the prolonged ray and it remains only to obtain one other point. Draw EN_1 parallel to OA. Since this is a central ray, it emerges from N_2 in a direction N_2D parallel to EN_1, and also to OA, since OA is likewise parallel to EN_1. As EN_1 is thus the central ray of a parallel pencil, the pencil will be focused at sone point on f_2. This point is evidently D, where the central ray cuts f_2 and accordingly BD is the emergent direction of the ray OA. In practice it is only necessary to draw N_2D parallel to OA, thus determining the point D.

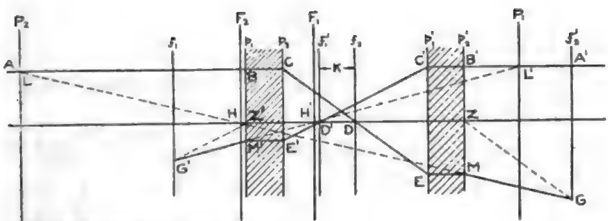
Similarly, the ray O_1A_1 will likewise pass between the planes parallel to the axis, as shown, and intersect the focal plane f_2 at D.

Fig. 11 shows a combination of two lenses. p_1 p_2 and f_1 f_2 are respectively the first and second principal planes and the first and second focal planes of the first lens. p_1^1 p_2^1 and f_1^1 f_2^1 are the corresponding planes of the second lens.

This combination may be represented by an equivalent lens. The focal planes and the principal planes of the equivalent

lens are obtained by tracing through the combination from each end, a ray entering the system parallel to the axis.

We know that the parallel ray AB from the front passes through the second focal plane of the equivalent lens. Tracing this ray through the two lenses by the Gauss method, the final direction is MG, which prolonged cuts the axis at H, and F_2 is the second focal plane of the equivalent lens; a farther prolongation cuts the original ray at L, and P_2 is the second principal plane of the equivalent lens.



ZG DRAWN PARALLEL TO C.E.
Z'G' " " " C'E'.

FIG. 11.

(An examination of Fig. 9 shows that in each case the principal plane lies at the point where the original parallel path of the ray is cut by the final path of the ray. This explanation will make clear the location of the principal planes in Fig. 11.)

F_1 and P_1, the first focal plane and first principal plane of the equivalent lens, are found in the same way by tracing the ray A^1B^1, etc., through the system.

The second focal plane (f_2) of the first lens overlaps the first focal plane (f_1^1) of the second lens.

Denote by K the distance between f_1^1 and f_2.

If F is the focal length of the equivalent lens and f and f^1 are the focal lengths of the two lenses, then

$$F = \frac{ff^1}{K}.$$

This equation is capable of mathematical proof, but the proof is not considered necessary in this article.

This formula holds good for any combination of lenses if we observe the conventions that K is positive when f_1^1 is in front of f_2 and that f and f^1 are positive when f_1 and f_1^1 are in front of p_1 and p_1^1 respectively.

F is a real quantity, although it may be either positive or negative.

The formula

$$F = \frac{ff^1}{K}$$

shows how the relative position of the two lenses affects the resultant equivalent focal length.

As the lenses are separated, K decreases, increasing F until when $K = O$, $F = \propto$.

A farther separation changes the sign of K, and F again appears, but with a decreasing negative value.

As above, when $K = O$, $F = \propto$, and pencils of light which enter the combination parallel emerge parallel. This is known as the telescopic condition, as it is found in all telescopes.

In connection with the above discussion of equivalent lenses the following should be borne in mind:

When two lenses having a common axis are separated by a finite distance it is impossible to find a single thin lens which, when placed in any fixed position, will produce an image of the same size and in the same position as that produced by the combination.

A single, thin lens can be found, however, which, when placed at a suitable fixed point, will produce an image of the same size, but not generally in the same position, as that produced by the combination. This is what is understood by an "equivalent lens."

ELEMENTARY TELESCOPE.

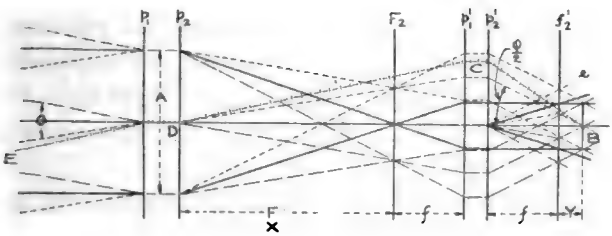
Fig. 12 shows two lenses fulfilling the telescopic condition, forming a simple astronomical telescope that produces an inverted image.

p_1 p_2 and p_1^1 p_2^1 are, respectively, the first and second principal planes of the first and second lenses.

F the focal length of the objective lens and f the focal length of the eye piece.

Then F_2 is the common focal plane of both lenses and f_2^1 the second focal plane of the second lens.

θ is the angle of the field.



$M = \frac{\varphi}{\theta} = \frac{F}{f} = \frac{A}{l}$

$XY = f^2$

$F = 2.25$ $\varphi = 60°$

$f = .75$ $\theta = 20°$

$X = 2.25$

$Y = .25$

$A = 1.50$

$l = 50$

WHILE "θ" AND "φ" ARE IN PROPORTION THEY ARE SHOWN EXAGGERATED AS "φ" CANNOT EXCEED 40°

FIG 12.

The paths of parallel pencils are traced out for the vertically-incident central pencil and for two pencils coming from the limits of the field. These three pencils focus as shown, then pass through the second lens and emerge parallel again.

From an examination of this drawing it would seem that the field θ could be enlarged by increasing the diameter of the eye lens (in such case the central ray from one edge of the field being the barred line B, C, D, E), but in practice, the eye piece is limited by spherical aberration to such a diameter that the ange φ is not greater than 40°. In practice, this angle is generally about 35°. A stop is fitted at F_2 of such a diameter that it cuts off all rays outside these limits. The objective is generally made larger than necessary because the lens is more or less imperfect near the edge. Consequently, a larger lens is used, then either partially covered or stopped down at F_2 until the clear aperture is of the proper diameter

to admit the light desired, and to limit the field to prevent spherical aberration.

The figure shows that all these parallel pencils form at e an image of the aperture A. This image is a bright spot that may be readily observed on a sheet of paper, held at the proper point, and is called the exit pupil.

To obtain the proper use of a telescope the eye should be at the spot e, for there the rays are parallel as the eye is accustomed to seeing, and the rays are gathered together in the smallest bundle. If the eye is positioned elsewhere, it is clearly apparent that there is loss both of light and of field.

The distance between e and the lens is known as the eye distance.

In such an elementary astronomical telescope an inverted real image is formed at F_2. This is magnified by the eye piece. The magnification of an object is evidently the ratio between the angle subtended at the eye by the final virtual image and by the object when seen by the naked eye. If we call the magnifying power M, it may be readily shown that

$$M = \frac{F}{f} = \frac{A}{e} = \frac{\varphi}{\theta}.$$

θ is the "apparent field" and

φ is the field of the eye piece.

The field of the eye piece is sometimes called the "true field." This is a confusing term.

As previously stated, the angle φ is limited to from 35° to 40°. Hence, it is apparent that $\theta = \frac{35^\circ}{M}$ and the field is absolutely fixed by the magnifying power.

The expression $M = \frac{F}{f}$ shows that we may increase the magnifying power either by increasing the focal length of the objective (F) or by decreasing the focal length of the eye piece (f).

The expression $M = \frac{A}{e}$ shows that if "e" is fixed at ″.25,

then A = M × ″.25, which gives the clear aperture required, which is less than the diameter of the objective lens. "*e*" always falls in rear of the second focal plane of the eye piece, as it is an image of the aperature of the first lens of this combination, and therefore its distance in rear of f_2 is obtained from the equation $XY = f^2$; where X is the distance of the aperature from the first focal plane of the second lens; Y the distance of "*e*" in rear of the second focal plane of the second lens; and f the focal length of the second lens.

The elementary astronomical or non-erecting telescope described above, consisting simply of an objective lens and a magnifying eye piece, is not suitable for naval purposes for the reasons that not only is the image inverted and reversed, but also that the movement of an object observed through such a telescope is reversed, so that an object moving to the right would appear to move to the left.

While gun pointers might possibly be trained to use such a telescope, it could only be achieved by long and arduous training, as it would be opposed by every natural tendency. Accordingly, we can immediately dismiss the thought of using this type of telescope.

It then becomes necessary to add an erecting system to the astronomical telescope with the disadvantage that each added lens uses up about 9° of the light which strikes it.

ELEMENTARY TERRESTRIAL TELESCOPE.

The elementary terrestrial telescope is obtained by adding an erecting lens to the simple astronomical telescope. The resulting lens system is shown in the accompanying Fig. 13,

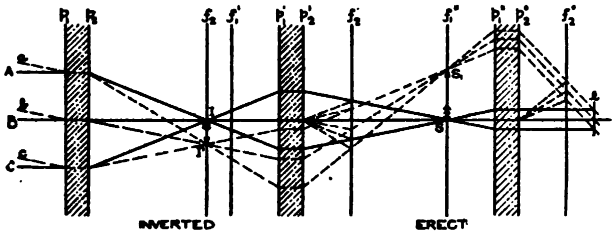


FIG. 13.

where three lenses are shown by the principal planes, $p_1 p_2$; $p_1' p_2'$; $p_1'' p_2''$. If A, B, C and a, b, c, are two parallel pencils radiating from the center and from the edge of a distant object, a real and inverted image will be formed as shown at II′ in the second focal plane of the objective lens. The only condition for the erecting system is that its first focal plane must be in rear of this inverted image. This erecting system or lens forms a real erect image at SS′. If this image is at the first focal plane of the eye piece, we have an elementary terrestrial telescope where the eye sees an enlarged image in its proper position. All that is necessary to convert this telescope into a gun-sighting telescope is to place cross wires either in the second focal plane of the objective lens, where the inverted image is formed, or in the first focal plane of the eye piece lens, where the erect image is formed.

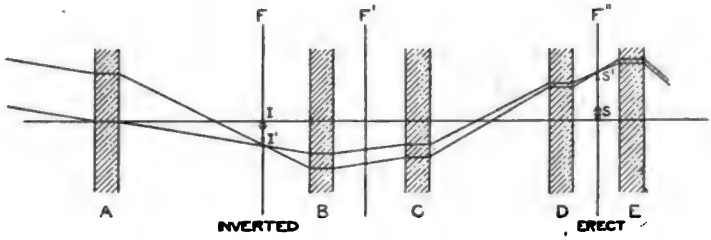


Fig. 14.

Fig. 14 shows the old Mark VII telescope. It consists of five lenses shown by their principal planes in the figures, lettered A, B, C, D and E.

A is the objective with focal length F.

Call the focal lengths of B, C, D and E f', f'', f''', f'''', respectively.

B, C, D and E, together are called the erecting eye piece; B and C form the erecting system; D and E form the astronomical eye piece.

B and C are so arranged that their focal planes overlap by by a distance K′.

D and E are similarly arranged, their focal planes over-

lapping by a distance K″, and the first equivalent focal planes (F′, F″) of both pair of lenses will consequently lie between them (see Fig. 11). Reference to Fig. 13 will show that B and C will perform the function of the erecting lens, as their first equivalent focal plane F′ is in rear of the image II′ formed by the objective.

B and C form an erect image at SS′. If this image is at the first equivalent focal plane of D and E, the telescope condition is fulfilled.

Assume

f as the equivalent focal strength of the system B, C, D and E,

f_1 as the equivalent focal length of B and C,

f_2 as the equivalent focal length of D and E, and

x the distance between the second equivalent focal plane of BC and the first equivalent focal plane of DE.

Then

$$f=\frac{f_1 f_2}{x};\ f_1=\frac{f'f''}{K'};\ \text{and}\ f_2=\frac{f'''f''''}{K''};$$

$$\therefore\ f=\frac{f'f''f'''f''''}{x\,K'K''};\ \text{but}\ M=\frac{F}{f};$$

$$\therefore\ M=\frac{FxK'K''}{f'f''f'''f''''}.$$

This equation for M shows the various ways of changing the magnification of the telescope. The magnification is increased by increasing F (using an objective of a longer focal length); or by increasing x, K′ or K″ (changing the position of the lenses); or by decreasing f', f'', f''' or f'''' (using eye-piece lenses of shorter focal lengths).

In all later telescopes the first equivalent focal plane of the eye piece, instead of lying between the eye-piece lenses as in Fig. 14, falls outside of the eye-piece lenses toward the objective lens, and the cross lines are accordingly moved forward to the same point.

FINAL VIRTUAL IMAGE.

(FIG. 15.)

As previously stated, the erect image formed in rear of the erecting system is real and can be seen on a screen placed at the proper point. This image is, however, very small and it is necessary to magnify it by means of the eye piece. The image actually seen by the observer is a virtual image of the erect real image and the cross lines. This virtual image is projected out in front of the eye piece at a distance depending upon certain conditions.

An examination of Fig. 13 in connection with the formula $XY = f^2$ will show that there are three cases possible in image formation by a positive eye piece, such as is used in erecting telescopes:

I. If the real erect image is in front of the first focal plane of the piece the emergent pencils are convergent, X is positive, Y is positive, and a real inverted image will be formed in rear of the second focal plane.

II. If the real erect image is in the first focal plane of the eye piece the emergent pencils are parallel, X is zero, Y is infinite, and a virtual erect image will be formed by the eye piece at infinity.

III. If the real erect image is in rear of the first focal plane of the eye piece the emergent pencils are divergent, X is negative, Y is negative, and a virtual erect image will be formed in front of the eye piece at the point where the rays intersect.

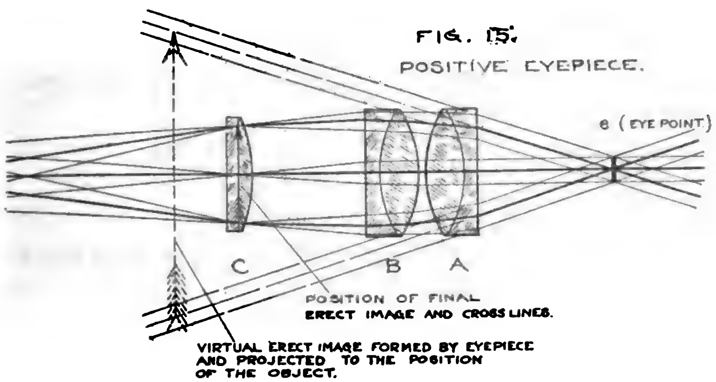
All erecting telescopes are found in case II or case III. Where conditions are as given in case III, the distance of the real erect image in rear of the first focal plane is so small that the virtual image will be formed at a great distance and, to the observer, will occupy practically the same position as the object itself; so that all the figures and formulae previously given will apply perfectly.

When an object is viewed through a telescope the final virtual image appears much nearer than the object; but, as

stated above, this is not the case, the final virtual image and the object occupying practically the same point. The apparent nearness of the image is an optical delusion, caused by the increase in detail due to the magnification.

THE FIELD LENS.

An examination of any modern telescope will usually show that the eye piece (Fig. 15) consists of three lenses, the two eye-piece doublets A and B, and the field lens, or eye-piece collective lens C.



The cross lines are etched by hydrofluoric acid on one of the inner faces of the compound lens C, then either silver and copper plated or filled in with a mixture of zinc sulphate and water glass, the two parts of the lens being cemented together with Damar balsam. In this way the cross lines are protected from injury and dirt.

It is apparent from Fig. 15 that when the pencils of light pass through the field lens C, a convergence is caused and the eye, situated in rear of A at the eye point *e*, will receive more of these pencils of light than would be the case if the field lens were not used. With the same diameter of eye piece, the use of the field lens enlarges the field of the telescope, hence its name.

PROOF OF THE PRINCIPAL PLANE THEOREM.

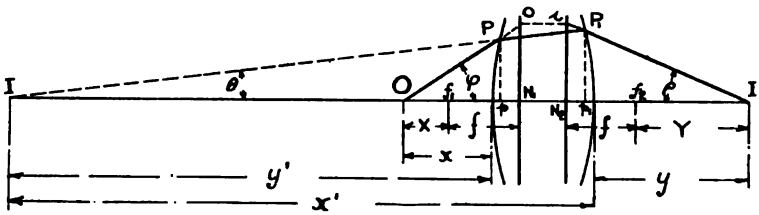


FIG. 16.

OP is a ray of a thin pencil of light from O, and P_1I is its path after emergence, forming the image I of the object O.

N_1 and N_2 are the principal points.

N_1o and N_2i are the principal planes.

The principal planes are cut at o and i by the prolongation of OP and IP_1.

f_1 and f_2 are the first and second principal foci.

To prove that $N_1o = N_2i$,

Draw Pp and P_1p_1 perpendicular to the axis.

As the pencil is thin, the distance from p and p_1 to the lens surfaces may be neglected.

I′ is the image of O by refraction at the first face.

The following equations have been given previously:

$$XY = f^2; \quad M = \frac{f}{X}; \quad M = \frac{yy'}{xx'}.$$

From Fig. 16,

$$\tan\varphi = \frac{Pp}{x}; \quad \tan\theta = \frac{Pp}{y^1} = \frac{P_1p_1}{x^1}; \quad \tan\rho = \frac{P_1p_1}{y}.$$

Equating value of Pp and P_1p_1,

$$x\tan\varphi = y'\tan\theta, \text{ and } x'\tan\theta = y\tan\rho;$$

$$\tan\theta = \frac{x\tan\varphi}{y'} = \frac{y\tan\rho}{x'};$$

$$\frac{x\tan\varphi}{y\tan\rho} = \frac{y'}{x'};$$

$$\frac{\tan\varphi}{\tan\rho} = \frac{yy'}{xx'} = M;$$

Also,

$$\tan\varphi = \frac{N_1 o}{X+f}, \text{ and } \tan\rho = \frac{N_2 i}{Y+f};$$

$$\frac{\tan\varphi}{\tan\rho} = \frac{N_1 o(Y+f)}{N_2 i(X+f)} = \frac{N_1 o\left(\frac{f^2}{X}+f\right)}{N_2 i\ (X+f)} = \frac{N_1 o(X+f)f}{N_2 i(X+f)X} = \frac{M(N_1 o)}{N_2 i}$$

But

$$\frac{\tan\varphi}{\tan\rho} = M,$$

$$\therefore M = \frac{M\ (N_1 o)}{N_2 i}.$$

Or

$$N_1 o = N_2 i.$$

EFFICIENT RADIO STATIONS.

By Richard Pfund.

With any given amount of power, it is not entirely the system, or the apparatus, that determines the transmitting efficiency of a radio station. It is its location and how the apparatus is installed that determines this, and particularly on land, and although the author's explanation may not agree with other theories, much experience and observation has nevertheless convinced him that the efficiency of such apparatus is most decidedly increased if its installation is carried out with a view to meeting the requirements indicated.

Briefly, two of the most important and most essential requirements for best efficiency, are, on one hand, as perfect a connection with the earth as a whole, or as large a portion of it as it is possible to obtain, and on the other hand, a suitable antenna, or elevated capacity area. As it is not always possible or practicable on land to obtain a sufficiently perfect connection with the earth by direct contact, a so-called counterpoise, or capacity earth, may have to be employed in order to, as it were, obtain a more effective grip upon as large a conducting area underneath as possible. The larger this area obtained, either by direct contact or inductively by means of a counterpoise, and the better and more uniform its conductivity, and the better in turn its contact with other and still larger and better conducting areas, the greater will be the efficiency of the station, and particularly when transmitting. As sea water is a very good conductor indeed, and as it forms the most extensive and most continuous conducting areas on the earth's surface, the most desirable location for a land station, under ordinary conditions, is at the seashore. Simply locating it near the sea, however, is not enough, but for best results it

must be located on a site between which and the sea a connection of low resistance already exists. With this end in view the station should be located at sea level, or below it, and, if possible, on marshy soil and as close to the sea as practicable. A quarter of a mile inland, or even on top of a low hill close to the sea, may make all the difference in the world on account of the character and extent of the intervening soil. In this connection the author vividly recalls the efforts that were made, and his practically complete failure, about eight years ago, to establish communication, with about 2 kw. at each end, between the old, and since abandoned, U. S. Navy Stations at Highlands, N. J., and Montauk Point, N. Y., both of which stations, although directly on the coast, were on top of dry hills. At about the same time, and with about the same power and equally efficient apparatus, the old Deforest Manhattan Beach Station, located in a salt marsh near the ocean at Coney Island, was breaking all previous records for range, and its high efficiency, as compared with other similar stations of that time, was no doubt due solely to its low resistance ground connection with the Atlantic Ocean, which the stations at Highlands and Montauk Point were unable to obtain on account of their location. The remarkable ranges obtained a little later, but still with comparatively low-power apparatus, at the U. S. Navy Stations at Fire Island, N. Y., Newport, R. I., Norfolk, Va., and Key West, Fla., on the Atlantic Coast, and at Mare Island, Cal., and other Pacific and Alaskan Coast Stations, were also no doubt due, not entirely to more efficient apparatus, but primarily to their having a more or less perfect connection with large bodies of sea water, *i. e.*, large continuous areas of high conductivity, and nothing else. The fact that it has at times been found possible to establish communication between stations on the Atlantic and the Pacific, afloat and ashore, is also no doubt due to the same cause, and if, for instance, the stations at Newport and Mare Island, which have at times been able to get in touch with each other with low-power apparatus, could each be moved say five hundred miles inland and there provided with the best

possible ground connection obtainable, they would undoubtedly find it very hard, if not absolutely impossible, although a thousand miles nearer to each other, to establish communication, or get a single signal to each other with the same power as before, and simply on account of their grounds being of very much smaller area and very much inferior in conductivity to those they have on the coast, resulting in a very much higher ground resistance between them. The further fact that stations on the Pacific and the Alaskan Coast and on the Pacific Ocean have been able to far outstrip stations of equal power and equipped with similar apparatus on the Atlantic Coast and Atlantic Ocean, and also the fact that the ranges obtained in the southern hemisphere exceed those obtained in the northern with equal power and similar apparatus, is also no doubt due to the same cause. A first-class low-resistance ground connection is therefore of prime importance in radio work, and this does not mean simply a first-class local ground of comparative small area, but, in the author's opinion, if two stations on the earth's surface are to communicate with each other most efficiently, there must be a low-resistance ground connection between them, and not only at each end with no connection whatever, or one of high resistance, in between. Otherwise the results are likely to be similar to those obtained when it was attempted to ground a telegraph wire in a bucket of earth in a balloon. By making connection with large areas of high conductivity at each end the chances of getting through with a given amount of power are very much improved, because, although there may be a large body of comparatively poorly-conducting material between, as, for instance, the North American continent between Newport and Mare Island, the actual ground resistance between these two points may be quite small, because this large body of poorly-conducting material is sandwiched in between, and is in intimate contact on both sides with two very large bodies of high conductivity—the Atlantic and Pacific Ocean—and then in addition, straight across the surface of the continent is not the only conducting path between them.

So much for the ground, or, as the author prefers to consider it, the ground return, and the next important matter to consider is the antenna, or elevated capacity area. For best efficiency in transmitting, an antenna should not only be as far away from the earth as possible, but it should also, for a given transmitter capacity, have the largest possible electrostatic capacity, with the shortest possible natural period and smallest possible resistance. This not only means that in order to most efficiently absorb and radiate the amount of energy supplied to it by a given transmitter, the antenna best suited for that particular transmitter must have a certain definite electrostatic capacity, but it also means that there is also, with the present type of stations, a certain definite best height above the ground for this antenna, because the higher an antenna the longer will be the leads to it and the greater will be the inductance and the resistance and the longer the period of the antenna as a whole, and this increased period due more to inductance than capacity. It is therefore obvious that the most efficient height of an antenna for a low-power set is very much lower than that of one for a high-power set and that we cannot, with the present type of stations, go on raising the height of the antenna, with a given size set, without materially impairing the efficiency. It is, of course, possible to reduce the inductance and the resistance of the antenna leads by increasing their number and also by properly spacing them with that end in view, but such an arrangement adds very little to the efficiency and, if carried too far, will actually decrease it, because the greater the number of leads and the more their spacing approaches, or exceeds, that of the antenna proper, and the nearer these spaced leads are brought to the ground, the more such an arrangement becomes practically equivalent to bringing the entire antenna nearer the ground. This, although it may reduce the inductance and the resistance and increase the electrostatic capacity, at the same time also reduces the efficiency of the antenna as a radiator. The reason for this may be clearer if we assume that the antenna and the ground underneath represent the two surfaces of a condenser—which

of course, they really are—connected to the terminals of the transmitter. The closer these two surfaces are to each other the shorter will be the connection between them and the greater will be the capacity, but at the same time the greater will also be the tendency of the oscillating charges to confine themselves in this condenser circuit, made up of the transmitter and the antenna and the ground underneath and the air dielectric between, and the less the tendency of the antenna end of this condenser circuit to radiate its charges into space, or, as the author prefers to consider it, the less will be the ability of the antenna end of this condenser circuit to act most effectively upon the upper conducting strata of the atmosphere, which, in his opinion, play a very important part indeed in the transmission of the energy between distant stations in which one side of the apparatus is connected to ground. The author knows of no formula for obtaining the best height of an antenna for a given amount of power with the present type of stations and believes that it can only be obtained experimentally. He feels certain, however, and not entirely on theoretical grounds, but also from actual experience, that with the following arrangement both the height of the antenna and the efficiency for all powers can be very materially increased. Instead of locating the transmitter on or close to the ground, and connecting it with the antenna by means of long leads, the transmitter is located at the top, or near the top, of, for instance, a substantial metallic tower of ample cross-section and surface, depending upon the size of set to be employed, and with the different sections carefully bonded, and with perhaps several additional conductors of maximum surface and low resistance in parallel with the tower structure, so as to reduce both the inductance and the resistance to a minimum. The antenna, if only one tower is used, would be of the umbrella type and supported in the usual way and carefully insulated from the tower. The latter would be carefully grounded, either direct, if conditions make this possible, or inductively by means of a counterpoise attached to it. In the latter case, of course the tower would

also have to be carefully insulated from the ground. The antenna would be directly connected, when sending, to one side of the transmitter at the top of the tower with the shortest possible lead, and the tower itself would be directly connected to the other side of the transmitter with a similar lead. With a so-called flat-top antenna—no doubt the most efficient—and requiring two or more supports, at least one of these supports would be in the form of a substantial metallic tower of the same general type as that proposed for the umbrella antenna. This tower would be located at that point at which it is desired to make connection with the antenna, and the transmitter would be located at the top, or near the top, and connected to the antenna and the tower in the same manner as with the umbrella antenna, and the tower would also be grounded, direct, or inductively, as before. It would not be necessary to also locate the receiver at the top of the tower with the transmitter, but the receiver could be located at or near the bottom of the tower, with suitable control apparatus for connecting and operating the transmitter and for disconnecting it and cutting in the receiver in its place when it is desired to receive.

It is, of course, obvious that in case a metallic tower is not desired, one of brick, stone, or other material, could also be employed, provided a sufficient number of leads for connecting the transmitter at the top with the ground, and of such shape, etc., as would reduce their inductance and resistance to a minimum, were also installed. The inductance and resistance of the tower structure, if of metal, together with the leads in parallel with it, or of such leads alone if a tower of non-conducting material is employed, cannot, of course, be indefinitely reduced, but this conducting path to ground will always add something to the period and the losses of the antenna circuit. On the other hand, however, such an arrangement of the transmitter will very materially increase the efficiency of the antenna as a radiator, or, as the author prefers to consider it, its ability to most effectively and most efficiently act upon the upper conducting strata of the

atmosphere, because by putting the transmitter in close proximity to the antenna itself, instead of on or near the ground, and then extending the antenna down to it, the antenna, together with the connection between it and the transmitter, is kept at the maximum distance from the surface of the ground. By this arrangement the losses due to a too-close approach between the antenna and the ground, *i.e.*, the two sides of the capacity in which the oscillations are produced by the transmitter, are limited to those between the tower end of the antenna and the top of the tower, or the leads to ground from that point, and are small on account of the very much reduced area of these grounded surfaces close to the antenna and the comparatively low potential between these adjacent portions of the ground leads and the antenna.

SUBMARINES—IMPROVEMENTS.

By Lieutenant (J. G.) C. N. Hinkamp, U. S. N., Member.

In a general review of the improvements in the submarines during the last fifteen years we find many things that are of great military importance, principal among which is the ability of the submarines to cruise long distances, and to make these cruises with such regularity that it no longer excites comment. Other items of importance are improvements in signaling and the ability to navigate with comparative ease under all conditions of weather. Habitability has improved, cruising radii have increased, submerged running has developed to an exact science, and in general it is evident that there has been constant advance in submarine effectiveness. This will continue as experience dictates what is and what is not necessary to the development of the boats for the service that they are to perform, namely, the discharging of a torpedo against an enemy.

ENGINES.

Submarines opened the way for large engines of the internal-combustion type, but the inherent features of submarine design demand absolutely reliable engines of light weight and high speed—a difficult combination to obtain. As the size of the boats increased the engine horsepower and size increased, and the requirements became more stringent. Designers had difficulty in proportioning the weight necessary for engine reliability and that necessary for the proper trim of the boat. However, the gasoline engine was brought to a high state of efficiency, and the boats that are now fitted with gasoline engines are able to operate as effectively as some of the more modern boats. Before any departure in design is accom-

plished a great deal of preliminary experimenting must be done, in order to determine the weak points of the new design and avoid them in construction. The departures that have been taken in submarine-boat engines are as follows:

1st. Gasoline engines, heavy type, "A" class.

2d. Gasoline engines, light type, "B," "C" and "D," "G-1," "G-2" and "G-4."

3d. Diesel engines, medium weight, four-cycle, "E" and "F."

4th. Diesel engines, light weight, two-cycle, "H," "K," "L," "M" and "G-3."

5th. Diesel engines, medium weight, two-cycle, double-acting, for seagoing submarines.

6th. Diesel engines, heavy weight, four-cycle and two-cycle, for coast-defense submarines.

Each step in the past was a distinct departure. By constant operation and study the defects in engines have been found and are being eliminated, so that in future boats we will be able to decide on a design of engine and feel fairly confident of successful results.

The auxiliaries of submarine engines have been a constant source of trouble, and on them dependence is placed for the solution of the heat problem. Another difficulty lies in the quality of the metal used in the various parts of the engine. In internal-combustion engines the initial pressure is very much greater than in steam engines and the mean effective pressure is but slightly greater. Metal suitable for use in internal-combustion engines is still in the process of development, and it is hoped that the day is not far off when the problem will be solved.

In comparing the submarine engines of our Navy with those of foreign navies, the fact must not be lost sight of that extraordinary cruising feats are the order of the day with a long coast line, while in the foreign navies operations from a base and short cruises are the usual order. When one hears of a long cruise made by foreign submarines the impression is

gained that our vessels cannot duplicate the performance. But if the situation is studied one finds that our vessels cruise more and farther as divisions than do most individual foreign boats, and with little or no publicity. This means that great advance has been made, but decidedly does not mean that excellence has been attained. There must be constant and hearty understanding and coöperation between designer and operator, the former being guided by the advice and experience of the latter, to the end that, within the limiting restrictions of policies and appropriations, each new craft will be an improvement on the last, and each step a step in advance.

MOTORS AND BATTERIES.

Advances in electrical engineering have been responsible for many improvements in submarines, especially in the motors, the storage batteries, and the controlling devices for these parts. The motors are built more ruggedly, are of the interpole type, and, generally speaking, can take care of all the current that the battery can supply.

The early control was simple, plain knife switches being used. The first improvement was the enclosing of these switches to minimize the danger from sparks that might ignite stray gasoline vapors. The next step was the contactor-type of control for the main motors. This permits of many variations in the location of the control, but has the disadvantage of being somewhat complex. However, for the large currents now handled it is necessary to have this type, and except for its complicated construction it is very much liked.

Batteries are of the lead-plate type. Great improvements have been made in this type of battery in the last few years, but the inherent defects of the type will never be overcome. The greatest disadvantage of the lead-plate battery is danger from chlorine gas. There is no such danger in the Edison battery, which as yet has not been used in submarines because of its initial cost. The rugged construction and the absence of

dangerous elements make it appear, however, that the Edison battery, in spite of its cost, is the cheapest in the long run.

NAVIGATING.

One is not impressed with the hardships of the submarine service by the sight of a present-day submarine with its large bridge and adequate protection from the weather. But a view of an older boat with a very low bridge and little or no protection against the weather, would impress one with the desirability of staying ashore on a cold night. The boats, when cleared for action, of course have no bridge on them. When cruising about on the surface, however, there is no reason why a few of the comforts which are possible on a large ship cannot be had. At best, the devices installed are but makeshifts for the real thing on board a large vessel, but their installation eliminates a great many of the earlier difficulties met in navigating the boats.

The older boats had compasses which were mounted in helmets and which were very difficult to read. Today the boats have not only the gyroscopic compass and standard compass, but a steering compass and, in some vessels, a compass mounted in the top of the periscope for submerged work. All of these additional installations are being specified and required as a result of experience in submarines gained by various officers of the Navy during the last twelve or fifteen years.

The gradual adoption of improved devices for navigation has greatly reduced the possibility of accident to submarines when cruising, both submerged and on the surface. The greatest step in advance is in the engine and motor control. This originally consisted of bell pulls and similar devices, which could not be depended on. Today, however, very reliable electrically-controlled indicators are installed. These, in addition to being connected to a large bell, are visual, such that the operator is required to look at the apparatus and read the order that is to be executed.

GYROSCOPIC COMPASSES.

The variations and deviations of a magnetic compass are not fully realized until they have been met on board a submarine boat. The hull is of magnetic material, and the necessity of having as few protuberances as possible makes it necessary for the compass to be mounted near the hull. In addition, there are large electrical currents always present, and in use under many different conditions. To obviate some of the difficulties, the compass is mounted in a helmet of composition. In order to get the reading of the compass card into the vessel so that the helmsman can read it, a reflector is fitted. In some of the boats, a compass was mounted in the top of the periscope, but it was not generally relied on. All this is changed in the later boats. Gyroscopic compasses are installed in all these boats, and the captain no longer has to worry about his deviations. Direction has become a thing of electrical engineering plus gravity, rather than magnetic directive force. In a well-adjusted compass all that is necessary is the setting of the speed indicator and the course. It is possible for the helmsman to point the boat by fractions of a degree now, where heretofore it was a question of quarter points. This means a great deal to the man who has to fire a torpedo at an enemy, for it is possible to do so without showing the periscopes, after once having obtained the enemy's bearings.

PUMPS, BALLAST TANKS, DRAINAGE.

Hydraulic engineering has advanced in strides along with the other branches of engineering—steam, electric and gasoline—and the effect of this advance has shown itself in the installations in the submarines. Considering all the ways that water has to be handled in the boats, greatest progress has been made in the pumping systems. The air blowing has not been changed materially and the principles have been applied in much the same way in all the types of boats; but great strides have been made with the pumps. The older pumps were of

the plunger type of limited capacity and, sometimes, doubtful efficiency. The rotary pumps that were installed were weak, of primitive design, indifferent workmanship, and subject to all the ills of this type of pump. Now the ballast pumps are of the centrifugal pattern, can handle vast volumes of water in a short time against a head of water double that possible ten years ago. The rotary pumps are rugged in design and of a material that will withstand the ravages of hard service. They are built with an idea of being lasting and serviceable. Impellers are so designed and built that they have ample bearing surface, with adequate supports. In the early boats there was considerable trouble with the shafting. It was of composition and broke very easily. Steel shafting was substituted. By this substitution the shafting became the strongest part of the pump, and eliminated most of the troubles of this type. Pumps of turbine design for the ballast pumps are being experimented with in some of the boats, and great results are expected, as it is claimed that they can pump against a head of water equal to the depth of test submergence. In the early boats, and some of the later ones, the circulating-water pumps for the engines were direct-connected to the engine, through gearing or chain drives. Invariably these systems gave trouble. The latest practice is to have all auxiliary and main pumps independently driven and so arranged that every reasonable contingency is provided for.

SPEED IN SUBMERGING.

One of the vital military requirements of a submarine is rapid submergence from cruising conditions. For cruising in the small boats it was necessary to rig a complicated bridge for the helmsman and other members of the crew. This took some time to unrig at sea when preparing to submerge. It was generally necessary to stow all the parts inside the boat through the conning tower because the seas were generally too high to permit of opening the main hatch. As the vessels in-

creased in size the freeboard became greater, the danger of shipping a sea much less, and it was more feasible to unrig at sea. But it was not until a year or two ago that the collapsible bridge was designed and installed. With this style of bridge it is only necessary to stow the stanchions inside the conning-tower fairwater, to fold the bridge deck down to the sides of the fairwater and secure them in the pockets provided for that purpose. It is possible to get the later boats submerged in a few minutes, because the normal amount of buoyancy is so great as to permit of the admission of a quantity of ballast while the preliminary work of unrigging is going on. Formerly it was necessary to allow about twenty minutes for this operation. It can now be done in from four to six minutes. With the bridge unrigged it takes from two to two and one-half minutes to get under the surface, traveling at three-fourths speed. Design, construction and training are responsible for the enormous reductions in the time necessary to perform any evolution.

RADIO, SIGNALS, COMMUNICATION.

Following the pace set in other fields of engineering, progress has been made in the methods of signaling. The contractors and the signal engineers in civil life have coördinated in such a manner that many improvements in this branch have been incorporated in the installations of later vessels.

The means of signaling on the surface are practically the same in the submarines as on the larger vessels of the Navy; namely, radio, flag, shape and sound in the day time, and radio, sound and light at night. Submerged, the signaling has taken tremendous strides. In the older boats there was no method of outside communication except that improvised by the crews, and the method was primitive. It consisted in tapping a rivet with a hammer and picking up the sound by holding the forehead against a frame. The introduction of the submarine bell opened the way for the later and better methods of signaling, all dependent upon the principle of setting up vi-

brations or waves in the water that are detected by means of microphones and heard through the ordinary telephone receiver. Today the submarine is well equipped to send or receive signals under all conditions, and the methods in use are vastly superior to those used only five years ago. The chief difficulty at present encountered is due to the internal noises of the boat. By the introduction of sound-proof compartments this bad feature can be eliminated.

PERISCOPES.

Improvements in periscopes have been very marked. The early periscopes were frail and leaky, and generally became cloudy after a short time. The periscopes of the present time, a few of which are in service, are rigid, clear and watertight, and the periscope tube is enclosed in an outer tube which has but one joint exposed to the weather. In the early periscopes the image inverted when looking to the rear with the eye piece stationary. In the present periscopes the image remains erect, no matter what the position of the object glass, and the bearing of the object is indicated by a movable pointer on a fixed dial. In addition to the feature of erect image in any position of the eye piece, the periscopes have been so arranged that a magnifier can easily be put into operation. This is of inestimable value when picking up objects. As soon as the object is picked up and easily distinguishable, a reversion to the normal eye piece can easily be made. Ordinarily the periscopes are used without the magnifier, in order that objects will appear as they would to the unaided eye, and also that distances will not be distorted. The monocular and binocular periscopes of the present day have all the advantages of the modern telescope or binocular glass.

ARMAMENT.

All submarines depend for their defense on their ability to submerge. Their armor is the water over them. Their offensive weapon is the torpedo. The methods of handling

the torpedoes has been improved, but even today the methods are crude, consume a long period of time and require, above all, entirely favorable conditions for comparative rapidity. However, progress is being made along well-defined lines, and there is hope that some day in the near future a means of handling the torpedoes will be devised that will not require "all hands" for the job.

The torpedoes are pointed at the target by pointing the boat, and their course is the boat's course at the instant of firing. The angle of the torpedo's course from the direct line—boat to target—is set and determined by means of the torpedo director. These instruments solve the angles automatically, working from certain well-known data, and if the torpedo is well adjusted it will hit either a moving or stationary target. The old method consisted in solving the angles mathematically and then taking a chance.

TORPEDO TUBES.

Nowhere were improvements more necessary than in the torpedo-tube arrangement. The older types of boats required that all the tubes be flooded every time that a torpedo was fired out of one tube. The tubes were covered by a cap which seated on the tube ends thus keeping them watertight. Now, by a door device, controlled from the inside of the boat, any or all tubes may be opened at the same time. The apparent advantages of the later arrangement are many. It is not necessary to pull into the boat, in order to avoid wetting it, a torpedo in a tube adjacent to the torpedo about to be fired. In torpedo practice this advantage is very apparent, as it is possible to work on torpedoes without interruption and fire practice shots at the same time.

LOADING TORPEDOES.

It would appear that, inasmuch as the submarine is designed for the sole purpose of firing torpedoes, adequate means would

have been provided for loading these torpedoes in the vessels. In the "A" class of submarines it was necessary to break the torpedo up into its component parts to get it into the boat. In the " B" class the torpedo hatch was located aft and in such an inaccessible position as to make loading of torpedoes a practical impossibility. Besides the danger of a passing wave washing water into the boat, there was always the difficulty of trying to bend the torpedo around a stanchion inside the boat. To overcome this difficulty the torpedoes were generally sucked in the torpedo tubes and as soon as they were in the tubes the cap was closed, torpedo tube drained, and the torpedoes hauled into the boat. Thus, before any work was done on them, the torpedoes were wet and often damaged by handling. In the " C" class of submarines, greater attention in design was paid to the methods of loading of torpedoes. They could easily be loaded at an average of about 20 minutes each. In the " D," " E" and succeeding classes the torpedoes can be loaded complete, but the best arrangements devised are crude.

It was a long time before proper and adequate cranes or davits were fitted to submarines for the handling of torpedoes. In the early period of development, it was necessary to parbuckle the torpedoes on to skids and load the torpedo into the boat from the tender, using a boat davit for this purpose. It is hoped that better installations for the handling of torpedoes will be designed. At best, torpedo handling is difficult in a submarine, owing to the peculiarities of construction not met in any other class of vessels.

FIRING TORPEDOES.

Torpedoes are fired by the old lanyard method. In the first boats the commanding officer gave the order to fire to the man at the tube; he controlled the firing entirely. Later the air-control was used. This control was mounted near the periscope, and the commanding officer himself fired the torpedoes when the boat was on the target. It is hoped that, in the near

future, the torpedo firing will be electrically controlled, in order to reduce, in a measure, the firing interval.

CONNING TOWER.

The early submarines of the "A" class have a protuberance on the deck called a conning tower. Compared with the later models it is entirely unsatisfactory. In later designs, the conning tower not only contains the auxiliary navigating gear, but is so designed and constructed that it can be used for an escape hatch for the crew in case of accident. The latest practice requires all the periscopes to be in the hull proper, the reasons being that this construction makes for greater safety, greater ease in the handling of the boat submerged and, if navigating the boat on the surface in rough weather from the periscopes, permits of three lookouts, instead of two, one man being in the conning tower.

TANK CONSTRUCTION.

The vessels of the present day have tanks which can withstand a pressure equal to that which the exterior hull can stand. In the earlier classes of vessels some of the tanks were so built that they could withstand a hydrostatic pressure of but from 10 to 50 pounds, and, when submerged, were always kept closed. The construction today permits of leaving the tanks open to the sea pressure without danger. In the construction of double-hull boats of various designs it is not necessary for the outer hull to be as strong as the inner hull, as it cannot be deformed, owing to the fact that it is completely filled with water. But the inner hull, or the hull proper, must be built to withstand the hydrostatic pressure of the specified depth to which the vessel is tested. Thus the present-day construction tends towards greater safety in the boats because the operators may safely put a high pressure on their tanks. With the sea valves closed, tanks can be pumped at any depth that the hull can stand.

CRUISING CONDITIONS.

The one great feature is the present-day submarine, which makes it possible to undertake long cruises, is the subdivision of the boat into compartments. In the "A," "B" and "E" classes there are no compartments, and the noise of the engines, which can be heard all over the boat at all times, is very loud and practically prevents any one from sleeping. Also, the one-compartment construction permits waste gases from the engine to permeate the entire atmosphere of the vessel, causing the men to become sick when any leaks occur about the machinery. The "C" class, after acceptance by the Government, were subdivided by a light bulkhead. In succeeding classes, "D," "F," "G," "H" and "K," the compartment construction was required, and the cruising success of present-day vessels is due largely to this construction. In addition to the comforts which are possible with this construction it makes for a degree of safety in the submarine which otherwise would be entirely absent. Any explosion or fire can be localized in the compartment where it originated and the crews can live safely in the other compartments. In case of a rupture to the outer skin of the ship it is possible to confine the flooding waters to a small area. The latest boats are so built that the flooding of one compartment will not completely sink the vessel.

HEATING.

To any one who has made a cruise up and down the coast in the winter time in a vessel without any heat the question of adequate heating appeals very strongly. The submarines of the "A" and "B" classes were not regularly provided with heaters, and great were the hardships encountered when cruising in northern waters. It was not until winter cruising became the rule rather than the exception that the idea of heaters was agitated to any extent. Electric heaters were at first supplied, but these absorbed large quantities of current from the storage battery. This reduced the submerged radius or re-

quired constant running of the engines to maintain the battery in a proper state of charge. The latest boats are fitted with steam heaters. These should eliminate many of the hardships attendant on winter cruising in northern waters.

HABITABILITY.

Going to sea in the early submarines is as near as one wants to approach to a real dog's life. Bridges are small, berthing of the men is difficult, there is little heat and the facilities for cooking are far from good. In the later boats these relics of barbarism have vanished to a large extent, and in their places are more modern equipment. The bridges are better, have more navigational facilities, and can be closed in for cold or extremely wet weather. In fact, cruising is more of a pleasure than it used to be, consequently the efficiency of the submarines is greatly promoted, and cruising efficiency is next in importance to the actual attack which is so dependent upon it. When the members of the crew know that they can turn in after a watch in a comfortable place, and feel assured of well cooked food, the general atmosphere is better. Contentment in this service, which requires the best efforts of the men at most unexpected times, and under very trying circumstances, must be most carefully considered. The advent of fuel oil as fuel for the engines, instead of gasoline, added largely to the safety of the vessel by reducing the danger from fire and explosion.

COOKING.

The comfort of the crew was not considered fifteen years ago. From the viewpoint of the present day, it appears that every discomfort which it was possible to conceive of was quite in order. The modern submarine has every comfort commensurate with the size and service of the vessel. The principal item making for comfort is, of course, properly-prepared food. The early boats used an alcohol stove, contrary to all rules and regulations for the safety of vessels burning

gasoline, but the men preferred to take this chance rather than have cold food. For this they cannot be blamed.

As time passed, electric cooking apparatus was installed. This was always subject to the many troubles inherent in early electrical heating apparatus. However, the idea was a step in advance. Today there is installed a well-arranged oven, four or five independent plates for cooking meats and vegetables, and an urn for keeping coffee constantly hot and on tap when cruising. All of these things, though small in themselves, make for contentment in the crew.

An important consideration for a long cruise is the stowing of adequate fresh provisions and an ample supply of dry food. The early boats had no means whatever of stowing food. What was not carried in cans was not carried at all. Now the vessels are provided with an ice-box large enough to stow three or four, and sometimes five days' supply of fresh meat and vegetables. The crew receive a ration equal to that served on our largest vessels. Dry food is stowed in specially-built lockers. The electric cooking, of course, is somewhat slower than cooking on a range fired by coal, but one man is detailed for this duty, and it keeps him busy. He is able to devote all his efforts to proper cooking. The present installations provide for all emergencies and are considered very good.

WATER STOWAGE.

Owing to the longer cruises taken by the boats, and the necessity for living on board many days at a time, the question of fresh-water stowage became an important one. The usual practice was to carry all the water the fresh-water tanks would hold, fill up the water breakers, hope that nothing would happen to the water, and trust that no one accidentally took a bath. The water was generally apportioned in such a manner that the crew were able to get two washes a day and all they needed to drink. This was very satisfactory. But to insure a ready supply, not only for the crews of the boats, but also for the

storage batteries, a small distilling plant is now installed that is capable of caring for the most extravagant needs of a crew. This may appear to be a needless refinement, but until the tenders for submarines are brought up to the standard that the service requires, and their number and their size is adequate, all these apparently extravagant mechanisms have to be installed. It is economy to have adequate tenders. Then many comforts could be obtained without the necessity of carrying so many auxiliaries on board the submarine, and the consequent saving in weight could be used in the design of more rugged and reliable engines.

SUBMARINE TENDERS.

The tenders for the submarines are inadequate. All of the tenders, so-called, except one, are make-shifts and, generally speaking, are not suited for the purpose. It is practically the only branch of the submarine service that has not shown a material improvement during the development of the submarine. There is only one submarine tender in commission that was designed as such, and another is building. Several are needed.

CONCLUSION.

Development has been conservative, improvements have been many, efficiency has been promoted, reliability has been increased, but the field is still open to advances, and the efforts of all should be bent to the establishment and execution of a safe and sane policy that will mean steady growth and progress in ALL branches of the submarine service.

ADDITIONAL NOTES ON SUBMARINES.

By George W. Baird, Rear Admiral, U. S. N. [Retired], Member.

As early as 1814 Robert Fulton wrote a letter to the President, Mr. Madison, on the subject of Torpedo Warfare and Submarines which, after a whole century, reads like a prophesy.

Mr. Fulton, in the letter, recites his experiments on submarine explosions, his floating mines, anchored torpedoes, submarine boat, etc., which seem to have anticipated the wonderful development of the submarine of today.

He met, naturally, earnest and determined opposition from naval officers in this country, in France and in England, who were probably wedded to the glorious and romantic traditions of the Service, as well as feeling an abhorrence of the ungallant and unseamanlike method of destruction of life and property.

The United States Navy was then controlled by a Board of Commissioners, with Commodore Rodgers as president and Captain Chauncey an active member, but subordinate to the Secretary of the Navy.

Congress appropriated $5,000 to defray the expenses of the Fulton experiments. Fulton sought to prove that gunpowder could be burnt under water with explosive violence; that cables (anchor cables were then made of rope) could be cut under water, leaving the ship to drift, and that torpedoes could be fastened to ships' sides with harpoons, fired from small arms, the harpoon being attached to the torpedo by a line; that torpedoes, on poles, could be thrust under ships and fired.

His plans were fully explained to the Commodore, a com-

mittee of Congress, and all others interested. The art of seamanship at that time was at its very highest, and American seamen were not excelled by any in the world, so that when the commanding officer of the *Argus* received an order to prepare a defence against a sham battle of torpedo attack, and knowing well the full particulars of the torpedoes and the method of attack, he knew at once just what to do. It was simple and was quickly executed. He surrounded the ship with his spare spars, held at a distance which exceeded the length of Fulton's poles, and dropped a netting from the water's edge to the bottom of the river, which were sufficient to defeat the operations of Fulton while the *Argus* was at anchor. This was made easier by the fact that Fulton had in his employ but a few men, and they were not seamen.

He did, however, succeed in cutting a cable under water and sending a ship adrift, and he proved that it was easy to explode gunpowder under water. He said that superior seamanship and previous knowledge of his methods, more than defects in his inventions, had foiled him in the North River experiments, and insisted that his devices in the hands of skilful operators would have succeeded. In this the Hon. Cadwalader D. Colder agreed, but the Commodore reported that "all that has been proven relative to this description of torpedo amounts to nothing when compared with the object for which it was constructed."

Fulton took his inventions to England, as many other Americans have done since then, and, in England he proved by test that ships could be destroyed from a distance, when he blew up the brig *Dorothea,* in October, 1805, in the presence of Mr. Pitt and Lord Melville.

The torpedo used was of the floating kind, which was carried by the tide to the brig, and exploded on contact. He repeated this in the presence of Admiral Holloway, Captain Owen and Captain Kingston, and quoted Captain Kingston as follows: "Twenty minutes before the *Dorothea* was blown up Captain Kingston asserted that if a torpedo were placed under

his cabin while he were at dinner he would feel no concern for the consequences."

Earl St. Vincent was quoted as saying "Pitt was the greatest fool that ever existed to encourage a mode of warfare which those who commanded the seas did not want and which, if successful, would deprive them of it."

Fulton had, before going to England, recommended that a large number of torpedo boats be substituted for a number of frigates, and showed by his figures how much less it would cost the nation. This was, probably, very provoking to the men "who commanded the seas," but, at the same time, it was out of the question for aggressive warfare, though the plan would be a good one for harbor defense.

Fulton's anchored (contact) torpedoes had the essential features of the modern mines, but lacked the size, high explosive power and mechanical construction which the state of the arts has now made possible.

Fulton declared that ships would never be sailed into harbors where torpedoes were anchored.

His submarine boat, the *Nautilus,* which he produced later was, in its essential features, the same as that of McClintock, shown on page 846, Vol. XIV.

Fulton was, however, apprehended by Bushnell, who was not only the pioneer of submarine-boat designers but invented the first screw propeller ever used, and for that boat.

David Bushnell, a recent graduate of Harvard University, was a native of Saybrook (now Westbrook), Connecticut, and, in 1775, entered upon mechanical pursuits and inventions.

The writer has made exhaustive searches for drawings or any trace of drawings or sketches of Bushnell's boat (the *Turtle*) without success, but from a description of the boat the writer has produced Plate I.

History shows that a Sergeant in the Revolutionary Army, named Ezra Lee—afterwards an Ensign—went down in the *Turtle* for the purpose of attacking one of the ships of the enemy. He reported that at a depth of three fathoms there

was light enough in the vessel to read by. He had 700 pounds of ballast, of which 200 were detachable at will, to increase the buoyancy of the vessel. His depth was determined by a syphon gage. His propulsion was hand-power, as shown. The torpedo, of the clockwork variety, was detachable (as shown), and attached by a line to a screw which was intended to be driven into the bottom of the ship of the enemy, as shown. He could propel the boat at the rate of three miles an hour for a short time.

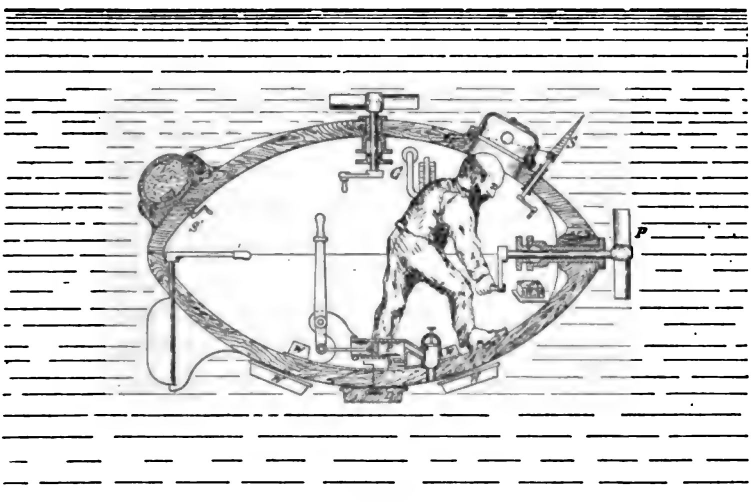


PLATE I.—BUSHNELL'S AMERICAN TURTLE, BUILT AND USED IN 1776.

He made but one attack. He reached the vessel, which was at anchor, only to find that he could not drive the gimlet-pointed screw through the copper sheathing of the ship. His torpedo became detached, floated down the river a short distance and exploded. This failure was simply for want of experience and for perfection of details.

In Vol. XIV, page 845, the writer has given a drawing and description of the famous submarine boat of McClintock which was used to destroy the *Housatonic*. In Vol. XXIV

the method of Col. Colt is also given for locating the position of a ship over a mine by a single observer on shore.

The periscope, now so successfully used in the submarine, and also the means of chemically purifying the air, are given, but the state of the art at that time had not reached mechanical propulsion under water.

The boat designed by Chief Engineer Wood and used so successfully by Lieutenant Cushing in the destruction of the *Albemarle* was essentially a surface boat, and has no place in the submarine class; but Mr. Wood's assistant (Second Assistant Engineer Lay) designed a very successful submarine, controlled from a distance, which was propelled by carbonic acid gas, carried in the vessel, and which Mr. Lay demonstrated as the material which would give the boat the greatest steaming endurance, but its radius of action was limited to the length of the electric cable which he used for the purposes of steering, firing, etc. Mr. Lay perfected many details, such as the steering gear, the vanes, etc.

After the dynamo electric machine was developed Mr. W. Scott Sims, of New Jersey, invented and built a boat which is manageable from a distance, and for which there is a field after these many years. His patents are dated in 1882. Unfortunately he did not live to promote them.

The vessel is shown in Plate II which is sufficiently clear to be understood by the readers of this journal.

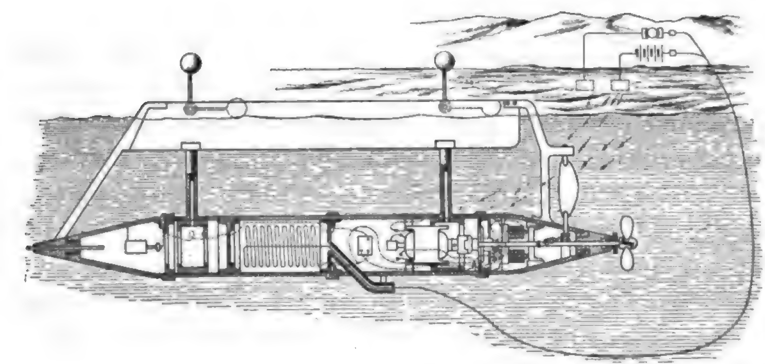


PLATE II.

The boat is propelled by an electric motor in the boat, the current being supplied by a machine on shore or on board a vessel, and its radius of action is limited to the length of the cable from the dynamo.

Mr. Sims found, as Fulton and McClintock had found, that it was difficult to keep his vessel on an even keel, no matter what her shape, when she was submerged and not in motion, and this was more difficult the sharper the model. Finally he made his vessel cylindrical, with the ends sharpened, and further attached a surface boat, as shown, with low freeboard, to keep the hull on even keel.

For sweeping a harbor of mines, with the least risk of life, the Sims boat, if used in pairs with a chain bridle between them, would be admirable. It would present a small target even when discovered, and if the surface boat, which is tied to the torpedo boat, as shown, were destroyed, the boat could easily be recovered by the cable.

The Sims boat is probably the least expensive to build of any of the automobile torpedoes.

NOTES.

THE FUTURE OF THE BATTLESHIP.

By Archibald Hurd.

On the very eve of the outbreak of war Admiral Sir Percy Scott stated that he could see "no use for a battleship and very little chance of employment for a fast cruiser" in future naval warfare. He announced that he had been driven to the conclusion that the submarine, with its menacing torpedo must be regarded as the supreme arbiter of naval supremacy. He looked forward to a complete change in the constitution of all navies: "Naval officers will no longer live on the sea, but either above it or under it, and the strain on their systems and nerves will be so great that a very lengthy period of service will not be advisable. It will be a navy of youth, for we shall require nothing but boldness and daring." He even went so far as to declare that the building of any more battleships in the present financial year would be "a misuse of money subscribed by the citizens for the defence of the Empire."

How far has the experience of the war sustained the contentions of this distinguished admiral, who has gained worldwide fame owing to his successful introduction of new methods for gaining the maximum fighting power from above-water vessels carrying the gun?

Every new instrument of warfare has a tendency to exert, for a time, an undue influence on public opinion, and sometimes also on naval opinion. When the automobile torpedo first made its appearance the First Lord of the Admiralty of that period announced that there was no intention of laying down any more battleships, as it was believed that the new weapon would drive the battleship, with its powerful guns, off the seas. At that time the submarine had not made its appearance, but considerable sensation had been created by the success of the French naval authorities in the construction of swift small surface vessels carrying the torpedo. In the course of various trials, which lacked reality, these little craft seemed to many observers to carry almost everything before them. The sensation and its influence upon constructional policy lasted for a few months, and then the inevitable awakening came and the building of battleships was once more resumed and continued with uninterrupted and increased vigor during later years.

The appearance of the submarine as a sea-going craft with great radius of action revived the belief in the superiority of the torpedo over the gun. Sir Percy Scott was not the first naval officer by any means to prophesy that the submarine would in future exercise primacy on the seas. As long ago as the spring of 1910, at a meeting of the Institution of Naval Architects, Rear Admiral Reginald Bacon read a paper on "The Battleship of the Future." He pointed out that "the problem of building a ship which cannot be sunk by the explosion of a torpedo is one that has exercised the skill of naval architects, and the design of a ship which shall not be incapacitated by such an attack has hitherto baffled all solution." Admiral Bacon did not go so far as to say that the battleship would become obsolete, but he urged certain conclusions of a somewhat revolutionary character as to changes in the type which met with considerable opposition from more conservative naval officers. Concluding his paper he said:

"All the considerations of offence and defence point to increase in size of battleships as modern gun construction advances. But since the modern battleship no longer holds the supreme position which, in the old days, made the battleship the sole ultimate arbiter of sea power, it is improbable that, as the torpedo improves, battleships, unable to defend themselves against any form of torpedo attack will be built merely to fight battleships.

"One function of the large cruiser will, therefore, be assumed by the battleship, high speed will become more and more necessary, and armor protection will be less accentuated than at present. The link between the ocean-going destroyer and the battleship will become closer, and one may reasonably expect that the huge monsters of the future will always be accompanied by torpedo craft of high sea-going speed as defensive and offensive satellites.

"The battleship, as now known, will probably develop from a single ship into a battle unit, consisting of a large armored cruiser with attendant torpedo craft. Line of battle, as we now know it, will be radically modified, and the fleet action of the future will, in course of time, develop into an aggregation of duels between opposing battle units. The tactics of such units open up a vista of most exhilarating speculation, and will afford to the naval officer of the future a scope for his tactical skill never dreamed of by us or our predecessors.

"The whole future is pregnant with radical obliteration of our present notions as regards tactics; but we may confidently prophesy that size of ships and power of gun will increase and increase until war, the great arbitrator among theories, will confirm or reconstitute our opinions regarding naval armaments."

Since Admiral Bacon wrote these words war has occurred. Popular imagination has been excited by the success of the submarine. No sooner had hostilities opened than a German submarine scored its first success by sinking the *Pathfinder*. Later on the three cruisers, *Aboukir, Cressy* and *Hogue* were destroyed by the same agency. Submarine attack was responsible for the sinking of the *Hermes* and the *Niger,* and in the Baltic the Russian cruiser *Pallada* fell a victim to the same type of ship. Moreover, it soon became common knowledge that German submarines had passed through the Straits of Dover into the English Channel and had cruised in far Scottish waters. Those who are unfamiliar with the development of the submarine were not a little surprised to learn that these vessels could proceed so far from their base, and it was thought that they must have some secret sources for the supply of fuel to enable them to travel 500 or 600 miles from the German coast. Those who had watched the progress of submarine construction were, however, in no way surprised by the activity of the enemy's under-water craft. They were aware that the submarine had been evolved into a vessel of considerable size, no mean habitability, and great radius of action. It was anticipated that during the course of the present war the submarine would exercise no little influence on the course of events.

The Board of Admiralty by the very success with which it mobilized the whole strength of the Royal Navy on the day preceding the declaration of war, saved the British Empire from many terrors and heavy losses, but at the same time conferred on the enemy the advantages of conditions favorable to the use of the submarine under the most favorable circumstances. By mobilizing the British Fleet before the German navy was ready, the Admiralty achieved two ends. In the first place they prevented war cruisers and converted merchant cruisers from escaping on to the trade routes, and Germany had in consequence to rely exclusively upon such vessels as happened to be at large when hostilities broke out. In the second place they robbed the German navy of the immense advantage of offensive initiative. It had been the deliberate policy of the Germans to

strike before our squadrons had been concentrated and had reached their war stations. In this the Germans were defeated. The British Fleet from the very opening of war converted the North Sea into practically a *mare clausam,* and sealed, to all intents and purposes, the two exits. The main naval operations of Germany were thus confined from the very opening of war to one of the most restricted oceans. The North Sea, considered in isolation, is a very large area of water, being of about 220,000 square miles in extent—in other words, it is two and a-half times the size of Great Britain. Considered, however, in relation to the other great seas of the world it is extremely small, since five-sevenths of the globe is covered by water. What British strategy effected was to confine the High Sea Fleet of Germany and its flotillas of destroyers and submarines to this area. The ships could escape only by accepting the challenge continually offered by the Grand Fleet under Admiral Sir John Jellicoe. This, on the one hand. On the other, these conditions offered to German submarines almost ideal opportunities for proving their usefulness, since the offensive-defensive policy of our Navy kept our forces concentrated within range of the enemy's submarines.

German submarine tactics have nevertheless so far proved a failure. No one would dare prophesy what the future course of the war may reveal, but a careful examination of all the evidence at present available shows that Germany's submarines have achieved no success which can influence the ultimate issue of the war, and that nothing has yet occurred to confirm the impression that under-water craft have rendered battleships and cruisers useless. Consider the leading events of the war. Did German submarines prevent a dozen large British ships invading the Bight of Heligoland and practically annihilating a considerable force of German cruisers and destroyers? Have German submarines prevented the ships of the Grand Fleet from sweeping the North Sea on more than one occasion, practically from end to end without loss? Have German submarines interfered with the most wonderful operation of which history contains any record, namely, the transport of British troops across the Channel, and the convoy to the Continent or to this country of tens of thousands of soldiers from the Dominions or from India, or have they succeeded in interfering with the general post which has been carried out in order to rearrange our garrisons overseas to fit in with the new strategical situation which came into view when war occurred? Have German submarines been able to save the German men-of-war, which were on duty in the outer seas, from being rounded up and destroyed? Have German submarines kept the seas for Germany's mercantile marine?

In each and every one of these respects the submarines of the enemy have failed. The successes which have been achieved have been achieved under favorable circumstances. The enemy's under-water craft have sunk a limited number of British ships, but they were slow ships, or at least were steaming slowly at the time of attack, and one of them was anchored when attacked. No submarine of any navy has yet been able successfully to attack battleship, battle cruiser, or armored cruiser when steaming even at her economical speed. The submarine, in fact, has been revealed as a type of ship which can be employed with deadly effect against large ships of deep draught when they are either at anchor or steaming slowly.

It is significant of the impression which the war has produced on expert minds in other countries, that on the other side of the Atlantic the victories which have been gained by German submarines have not yet affected constructional policy. In its last session Congress authorized the building of three of the largest battleships ever designed. These ships had not been begun at the time when war occurred. Considerable curiosity was felt as to what action the Navy Department of the United States would take in view of the early successes of the under-water craft in the

North Sea. It was assumed that the American naval experts would not ignore the progress of events in Europe. It soon became known that they were indeed watching matters with the closest possible attention. Week succeeded week and no action was taken, but at length it was announced that it had been determined to proceed with the construction of these three ships. This announcement was followed by the publication of a memorandum which put on definite record the conclusions which the Department had reached as to the limited influence of the submarine.

In accordance with these views not only have the keels been laid down of three battleships, which are of 32,000 tons displacement, but Congress has been recommended in its present session to authorize the construction of two more of these large units. It is thus apparent that the naval authorities of the country which, when the war opened, was running a neck-to-neck race with Germany for the position of second greatest naval power, have no intention of abandoning the construction of large-gun ships. They still believe that the battleship holds its traditional position as the ultimate arbiter of sea power and it is intended to pursue an active policy of construction.

The most remarkable fact revealed in this connection is that the Americans still persist in refusing to build battle cruisers, and are content with vessels with a speed of only 21 knots. It was reported early in last year that the General Naval Board favored an advance of speed to 25 knots, and the limitation of the number of guns to eight of an increased caliber, going from 14 to 16 inches. A vessel of this type would have been an improvement on the *Queen Elizabeth* class, and would have involved a very great increase in individual cost. It is understood that the Navy Department viewed this departure with no favor. Consequently the American Navy is being provided with armored ships deficient in speed in comparison with many vessels which have been constructed for the navies of European Powers, and at the same time no effort is being made to build any swift ships ranking in size between the battleship and the destroyer. No cruiser of any kind has been laid down for the American Navy since the three vessels of the *Birmingham* class were launched in 1907. The result of this policy is becoming apparent. Next spring the United States will possess only 13 cruisers of less than fifteen years of age and displacing 6,000 tons, and will have only nine small cruisers. Most of these ships are becoming obsolete and are relatively slow, and yet the new shipbuilding program shows that there is no intention of making good the growing deficiency in cruising ships, and expert authority has refused to increase the speed of American battleships. Of course the strategical conditions in American waters are radically different from those which affect the policy of European Powers, but nevertheless this continual neglect of cruiser types and contempt for the value of speed in large armored units must cause some surprise.

It may be assumed on all the evidence which is available that the battleship will continue to exercise primacy on the seas and that, in spite of the policy which the Americans are pursuing, cruisers in considerable numbers will continue to be built by the leading naval Powers. But the battleship of the future will undoubtedly differ very considerably from the battleship of the present. Of all the revelations which the war has made none has been so remarkable and awe inspiring as the ease with which a well-aimed torpedo can destroy the largest of vessels. One of the cruisers of the *Cressy* type was sent to the bottom by a single torpedo. She was designed before radical changes were made in hull construction, owing to the improvement of the automobile torpedo, and there is reason to anticipate that a large vessel of more modern design would not have been destroyed at a single blow. But, however this may be, the indubitable fact remains that surface ships of war have not resisted explosion as it was thought that they would resist it. Every naval designer

and constructor will admit that the precautions which have hitherto been taken against mine and torpedo have proved inadequate.

It is already apparent that the battleship of the future will have to be designed on new lines. It may be that even the suggestions which are put forward by Mr. T. G. Owens in the paper which was published in "Cassier's Engineering Monthly" of May, 1914, will prove inadequate. It is possible that, while not abandoning the system of elaborate subdivision of bulkheads, some other means will have to be found for protecting the battleship against the torpedo, whether fired by submarine or destroyer.

In this connection it is interesting to recall that in the "United Service Magazine" of June, 1910, "Master Mariner" put forward a suggestion for rendering the battleship less liable to destruction from explosion. He controverted the opinion that it was impossible to produce a satisfactory form of external passive protection to a ship. He declared:

"We have for years accepted the general principle of such a defence in a half-hearted way, by providing all our large armored ships with a clumsy movable arrangement of nets, which give a very indifferent protection at the cost of effectually hampering all movement on the part of the ship. At that point we have stuck, save for minor improvements in the details of construction of the nets themselves and their appurtenances. But poor as this arrangement is, it was found sufficiently useful under the stress of war by Russia and Japan to retain its place in the equipment of their navies, and it is surely worth our while to follow up this lead experimentally at least. The main idea underlying any external defence evolved on this principle is not to resist the force of the explosion of a torpedo at the point of impact, but to *cause the explosion to take place at a sufficient distance from the hull of the ship to render it ineffective as regards serious injury to her buoyancy or her machinery*.

"If this idea is to be followed up to any purpose, however, we must be prepared for radical innovations in the external fittings—or even structure—of our ships, and it is just this point which largely accounts for our lack of progress. Any proposal which smacks of 'revolutionary' changes is regarded with suspicion in the Navy, and always has been. This extreme conservatism exists in all matters connected with the service, but perhaps in none so much as in those which affect naval ideas as to what properly constitutes a 'ship.' It is an historical fact that a picked committee of experienced flag officers and captains once officially condemned the introduction of steam as a most dangerous innovation in vessels of war. We should not be justified in blaming them in the light of our later knowledge, for the steam engine of those days was a very different affair to that of our own, and the views of these officers were colored—or, more strictly speaking, completely dominated—by the training of their whole lives to work under and accept conditions which this 'dangerous innovation' would undoubtedly change. But if their report had received the concurrence of the authorities the propelling power now in such effective use would never have been attained, except through the practical demonstration of its value in the Merchant Service and foreign navies. Similarly, the late Captain Cowper Coles—a man far ahead of his contemporaries in foresight and independence of judgment—had the utmost difficulty in persuading the Admiralty to convert the three-decker, *Royal Sovereign*, when on the stocks, into the radically different, but infinitely more serviceable, turret ship which she eventually became. In this instance it actually was the example of foreign countries which decided the Admiralty to act upon his suggestions. Cases such as these—and their name is legion—are instructive in proving that the common disinclination of the service to give due consideration to an entirely new idea, from an impartial standpoint, is by no means always justified.

"The innovations necessary to provide a ship with a complete and

efficient form of external defence against torpedoes would not involve, by any means, so drastic a change as the alterations whereby the *Royal Sovereign* was converted from an unprotected three-decker, carrying 100 guns, into an armored shell-proof turret ship carrying ten. But half measures would be a mistake, nevertheless, and no arrangement of nets dangling from the ends of swinging booms would mark a satisfactory advance, however great an improvement on our present equipment of that kind. The point to be realized is that to afford a real and durable protection under all conditions, an external defence would have to take the shape of a fixed and permanent structural addition to the ship. If we are prepared to accept the undoubted drawbacks and handicaps it would impose for the sake of the immense boon conferred in other directions, the provision of such a defence offers no insuperable difficulty. A screen of steel plates, slats, or bars answering the purpose could be devised without calling for any exceptional effort of constructive genius, and the security provided by some such arrangement would very amply counterbalance the disadvantages inseparable from its use. With our ideas wholly or mainly shaped by constant association with the ships of the present day, it is perhaps difficult for us to assimilate the notion of a ship with such an incumbrance surrounding her. But, in truth, the effort of imagination required is as nothing to that which the sailor of the early or middle nineteenth century would have been called upon to exert to foresee a *Dreadnought*, or even a *Devastation*."

"Master Mariner" admitted that such a form of defence was not unaccompanied by drawbacks, but he held that the advantages that it would offer were too apparent to call for strenuous advocacy.

"The Capital Ship would at once resume the place held by her predecessor of a hundred years ago as Queen of the Seas, fearing nothing afloat by day or night, moving at will undeterred by any unseen danger. And the drawbacks themselves are on the surface, as it were, and sufficiently obvious to be foreseen and dealt with as far as practicable in advance. First of these is the docking question. A ship with a fixed screen would, of course, possess a much greater beam than one without, and require wider docks than any we possess. These would have to be built, unless open slipways were found preferable. And as the screen would prevent the use of docking shores, the ship herself would need to be constructed with a rectangular midship section to rest without shores on a "gridiron" dock, as flat-bottom steamers do in many parts of the world today. Then the screen would add, perhaps, a thousand or more tons to the weight a vessel would have to carry. Increased size of hull would meet this, and the advantage of a torpedo-proof defence would justify its existence, even if it weighed 20 per cent. of the ship's whole displacement. A reduction of her speed would be another unavoidable consequence. To what extent this would follow it is very difficult to foretell, but if the screen had openings allowing the free passage of the water in the fore-and-aft line—though too narrow to admit a torpedo—the loss of speed might not be very great. In any case this also could be met by increased size of hull allowing for greater engine power.

"Increased size of hull and engines would involve increased cost in the individual ship, no doubt, apart from the expense of the screen itself. But it would not involve an increase in the naval estimates as a whole. If we can produce ships to which the threat of the torpedo means little or nothing we shall be in a position to effect a very material reduction in the number of destroyers we lay down every year. Our destroyer flotillas are partly intended to safeguard our larger ships, by providing a force to attack hostile torpedo craft, and Admiral Bacon thinks that this reason for their existence will be emphasized in the future. If hostile torpedo craft are no longer to be feared, the necessity for this form of protection will vanish, and we shall need only to build such destroyers as are

required for purely offensive operations. Any great reduction in our annual output of destroyers would quite counterbalance the extra cost of screen-protected battleships."

The writer did not profess to offer proposals for a detailed design of the screen, admitting that that would fall within the scope of the naval constructor, but he, on the other hand, indicated what he regarded as essential.

"In the first place, then, it would have to be stout enough, as regards framework, to stand any kind of weather. To minimize the effect of pressures set up by movements of the ship, or the action of the waves, a cagelike structure of steel bars or slats would offer the advantages of reduced area of resistance if it could be made sufficiently strong to catch and hold, or explode, a torpedo. Any form of screen would, moreover, require to be quite independent of above-water fittings or supports liable to damage by gun-fire if it was to remain an effective protection after an action with other battleships. This, however, offers no great constructive difficulty. To extend the screen forward, ahead of the stem, would probably be superfluous, if it was carried into the stem itself from the supports on either bow, and the stem and forward compartments were strengthened. A torpedo striking the defence very far forward would, in such a case, explode comparatively near the shell of the ship no doubt, but opposite a small compartment, where the framework was specially constructed to minimize its effects. It would be essential also, to ensure that the screen was so constructed that no torpedoes could get behind it if fired from astern, as they sometimes do with any system of mere beam defence. Yet another important requirement would be that the screen should reduce the speed of the ship as little as possible. Generally speaking, an arrangement of horizontal bars would offer the least resistance, either direct or frictional, to forward movement, but experience would be necessary on this point before the best designs could be definitely ascertained. Lastly, the framework of the screen would have to be strong enough to allow the vessel to rest against it when lying alongside a jetty.

"It is not suggested, of course, that any form of screen strong enough to withstand the explosion of a torpedo altogether as regards damage to itself could be designed within practical limits as to weight. But it might be designed so that the damage was well localized and small as to general results. That is the case with the steel wire-net defences now in use, and is one of their few good points. If a hole was blown in a plate, or a few bars fractured—according as the screen was made of one or the other—the probability of a second torpedo finding its way exactly through the hole made by the first would be merely one of the very remote risks inseparable from the use of any design of war material intended to act as a shield.

"It may, perhaps, be argued that the introduction of such a form of protection would be immediately countered by an increase in the size of the torpedo, and very possibly, no doubt, that might be one of its first effects. But to be of any use the increase would require to be pronounced—in itself a great gain from a defensive standpoint. An increase would add greatly to the difficulty of handling torpedoes in destroyers and submarines, and reduce the numbers of the many torpedo craft, large or small, could carry. Torpedo craft themselves would then require to be bigger, and therefore fewer in number for a given financial outlay, and more easily sighted and dealt with by gunfire at night. We might, indeed, perhaps witness as a result of the introduction of defence screens a parallel to the enormous increase in the size of guns forced upon all navies by the introduction of armor. It may be that in this latter contest the gun is at last the victor, as Admiral Bacon thinks, but even if that is so, armor has immensely added to the difficulties with which the gun has had to contend, and severely limited the number of effective guns a ship of the line can carry."

In the light of what has occurred during the present war, all who are concerned in the creation of naval power will be interested to read again the suggestions of this "practical seaman," as he described himself.

It may be that the battleship of the future will be provided with some such screen as this writer has indicated; the war has certainly shown that an external defence of this character would be a great source of strength. It may be that, in accordance with American policy, each unit will be much bigger than anything hitherto known in Europe. It is probable that there will be considerable development of the scheme for the subdivision of the hull, and it is possible that the ship will be provided with a bow somewhat resembling that which has been given to the newest Russian battleships. This would strengthen the large ship if at any time she had to ram an enemy's larger seagoing submarine when running on the surface. It is probable also that the battleship of the future will be almost as swift as the vessels which are now known as battle cruisers, since speed and the use of the helm have been shown to be the best defence against the torpedo or the submarine. In any case we may regard one thing as certain, and that is that naval constructors will put up a strenuous fight for the survival of the large-gun ship, with its suggestion of power so patent in peace, when diplomacy fights our battles, and there can be little doubt that as a result of their labors the battleship will remain the supreme unit of naval power.—"Cassier's Engineering Monthly."

THE APPLICABILITY OF ELECTRICAL PROPULSION TO BATTLESHIPS, TOGETHER WITH THE EXPERIENCE GAINED WITH IT ON THE *JUPITER*.

By Lieutenant S. M. Robinson, U. S. N.

ABSTRACT.

The object of this paper is to show the best method of ship propulsion to be applied to battleships. It has been attempted to keep out all matters that do not bear directly on this one point. The *Jupiter* has been introduced merely to show the reliability of the apparatus that it is proposed to use. The Navy Department has authorized an electrical installation for the new battleship *California*, and it is believed that she will mark the beginning of a new era in marine engineering.

Briefly, the machinery for a battleship would consist of two turbo-generators running at a maximum speed of about 2,000 revolutions and generating current at about 3,000 volts, two switchboards, and four induction motors. The latter would be arranged with one on each of four shafts. The motors would be of the double squirrel-cage type and the stators would be fitted with pole changers which would give the motors two different numbers of poles and consequently two speed reductions. The motors would have two independent squirrel-cage windings, the outer one having conductors of high resistance and the inner one conductors of low resistance. At high frequencies, such as would obtain in the rotor when starting up or backing, the inner winding takes practically no current due to its high resistance under these conditions; therefore the high-resistance winding on the outside is the only one in operation and this insures a large torque for starting or backing. When running close to synchronous speed, the resistance of the inner winding is much reduced and it then operates like any ordinary squirrel-cage motor.

In choosing a method of propulsion for battleships, there are six points to be considered. They are, in order of their importance: (1) Reliability, (2) maneuvering qualities, (3) economy, (4) space occupied, (5) weight, (6) care and upkeep.

RELIABILITY.

Taking these points in their order, the first to be considered would be reliability. The reliability of induction motors and large high-speed turbo-generators for land use has been well known for some time, but up to about one and one-half years ago had not been demonstrated on board ship. However, experience at sea with them has not developed any trouble, and their reliability is now unquestioned. There are several things that go to make an electric installation more reliable than any other. First, the installation is in duplicate throughout and the breaking down of one engine does not affect the ship except at high speed; it might be said that this is true of other installations, but it is not true in the same sense nor in the same degree that it is with the electric drive. For example, if one turbo-generator breaks down, the ship can still run in a perfectly normal manner up to a speed of about 19 knots. This would be impossible with any other mode of propulsion; even at speeds within the power of one engine on a twin-screw ship the maneuvering qualities would become so bad as to handicap the ship; in fact, a twin-screw ship operating with one engine would be able to reach port safely but would be of very little use in battle, whereas an electrically-driven ship, operating with one turbine, would be just as good as any ship up to a speed of about 19 knots.

Then, too, there is the question of auxiliaries; with other forms of propulsion, if the auxiliaries in one engine room break down, that engine room may be almost entirely put out of commission. Of course the main air and circulating pumps are cross-connected, but when operating that way they are not very satisfactory, and I have seen a battleship forced to stop because the forced-lubrication pumps in the starboard engine room broke down. If the ship had gone ahead with the port turbines, the starboard turbines would have revolved and burned out the bearings, so that it was necessary to stop the ship and repair pumps; with electric drive it would be a simple matter to shift to the other engine room and make repairs without stopping.

There is one thing that makes the installation more reliable than other forms of turbine drive, and that is the fact that the turbine revolves in the same direction all the time. The importance of this cannot be overestimated, as it is believed by operating engineers that the greater amount of blading trouble that ships have had is nearly always due to the distortion that occurs when backing. There is also one other advantage when compared with reciprocating engines, and that is that the turbines would be less susceptible to damage by water when the boilers are priming. This fact has been thoroughly demonstrated by experience on the *Jupiter.*

MANEUVERING.

As regards maneuvering qualities, it is believed that the electric drive is far superior to any other method. This point will be taken up more in detail when the *Jupiter's* installation is discussed. It is rather difficult to describe these advantages on paper, but very easy to appreciate when you see the machinery working. Instead of big, heavy throttles to open and close, there are light, easily-handled oil switches, and a speed controller that can be handled with one finger. The engine-room watch required for a battleship would be just half what it would be with other means of propulsion, as one engine room would be idle practically all the time. In a rough sea there is no racing, with its attendant strains on machinery and on the personnel on watch. Any desired speed can be very quickly attained. The speed can be very accurately maintained and without any effort on the part of the personnel. These last two points are of very great importance on battleships when maneuvering in forma-

tion. The stand-by qualities are also superior to those of other installations, for at a very small expense in the way of steam the main turbine can be kept running very slowly and the engine room is then ready to answer signals at any time; this might be of great importance to a battleship. The backing qualities of this installation are also superior to other forms, particularly turbine installations. It is possible to attain full power in the astern direction.

ECONOMY.

In considering the question of economy, we have reached the point which first suggested the use of electricity for the purpose of propulsion. None of the older forms of propulsion, such as reciprocating engines or direct-connected turbines, can compare at all favorably with this method, the gain in economy being over 20 per cent. at high and cruising speeds. The only other method of propulsion that would appear to compete with this method in economy would seem to be the combination of high-speed turbines and reduction gears. The loss in the latter amounts to about 2 per cent., while in the electric-reduction gear the losses are from 8 to 9 per cent. Mechanical reduction gears have been developed to a high degree, and it is very probable that in quite a number of cases the mechanical gears would be preferable to the electric drive, but as this paper is limited to a discussion of battleships that subject will not be touched.

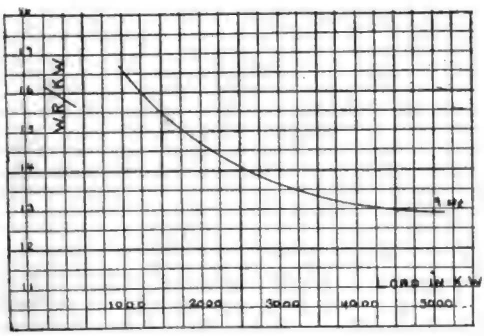


FIG. 1.—4,000 K.W. TURBINE, U. S. S. JUPITER.

For battleships I do not believe there is any question of the relative economy of the two methods—the electric drive is far superior. The electric reduction method possesses two inherent advantages that the mechanical method cannot overcome. First, the electric installation uses only one turbine at low powers; second, the induction motors are fitted with pole changers which allow the turbine to be run at normal speed with the ship cruising at low speed—in other words, the electric drive permits of two speed reductions while the mechanical gear has only one. These two advantages exist no matter how the two installations may be laid out, and they far outweigh the difference in the losses of the two methods. To show just how great these advantages are, two curves are shown. Fig. 1 shows the *Jupiter's* turbine operating under varying loads and Fig. 2 shows the turbine operating under varying speeds (and also loads). The load curve does not really show how very bad the conditions are for a battleship, as in that case the power at 12 knots is only about one-seventh what it is at 21 knots and the curve shown does not give so great a per cent. of reduction. It will also be seen that the speed curve is very steep at the low speeds and shows the bad effect of reducing tur-

bine speed very greatly. As nearly all the cruising of a battleship is done at fairly low speeds, it is evident that the electric drive will be more economical than the mechanical gear.

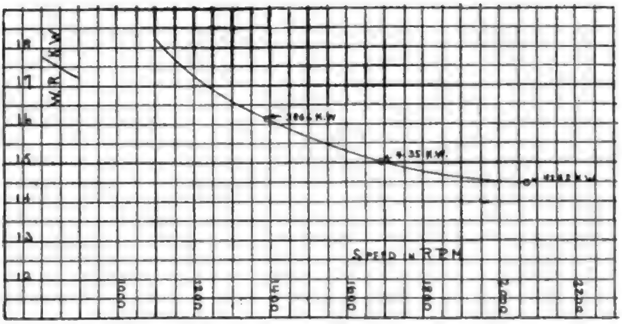


FIG. 2.—4,000 K.W. TURBINE U. S. S. JUPITER.

In addition to the two advantages just stated, there are a number of other things entering into a comparison of the two methods that make it very doubtful if the mechanical gear would be as good, even at the high speeds, as the electric reduction. It would probably be necessary to use at least four turbines with a mechanical gear instead of two, as with the electric method; this fact alone would make the electric reduction the more efficient of the two. Then there are also the very considerable losses due to the friction and windage of the backing turbine. According to Sir Charles Parsons this loss amounts to about one-half a per cent. Also there would be a saving due to the fact that only one set of auxiliaries would be used at a time.

SPACE OCCUPIED.

As regards the actual space that the machinery would occupy, the electric drive would take up less space than any other installation, with the possible exception of the mechanical gear. The arrangement of the machinery, however, is so much more flexible with the electric drive than with other methods that it is probable it would occupy less space than would the gears. It lends itself very readily to watertight subdivision.

WEIGHT.

The next point to be considered is that of weight, and here again the electric drive is superior to any other method with the possible exception of mechanical gears. It is rather difficult to say exactly how the two methods would compare either as regards space or weight, as a great deal would depend on the number of turbines used with the mechanical gear in order to get 30,000 shaft horsepower. However, if the largest sizes of geared turbines now at sea are any guide, the weight of the electric drive would be less than the weight of the geared turbine.

CARE AND UPKEEP.

As regards care and upkeep, the electric drive is greatly superior to either reciprocating engines or direct-connected turbines, also due to the fact that the turbines do not have to reverse, it would be superior to the geared drive in that it would have less blading trouble. Either of the two latter methods, however, is very satisfactory in this respect, as a high-

speed turbine is so small that it is easily handled and repaired by the ship's mechanics.

This completes the case of the electric drive for battleships; all of the points have been considered and in every case it has been seen that it has no superior. There are, however, some points that have been made against electric propulsion, and these will be taken up before proceeding to a discussion of the *Jupiter's* installation. First, there is the danger from large quantities of water in the engine room; however, this danger is more fancied than real, as all the wiring could be placed well overhead, being taken out of the tops of the motors and generators. The generators could be placed high enough to insure their safety and the motors could be placed in watertight pits so that the engine room could have enough water in it to put the auxiliaries out of commission before reaching the main engines. The next point is that, when operating with one turbine, all motors must run at the same speed if they run at all; this fact has been found to be no handicap in actual operation. The next point is that, in turning, the inboard screw does not slow down; this slightly increases the turning circle, but does not increase the space necessary to turn in, as the inboard screw can always be stopped or backed.

The equipment on the *Jupiter* is similar to that which would be used on a battleship, so it offers a good example of many points that have been advanced in favor of the electric drive.

U. S. S. "JUPITER."

The *Jupiter* has been in commission over a year and a half. During that time she has conducted two official trials and has carried on the usual routine work of a navy collier. She has steamed about 14,000 miles and has been handled a great deal around docks, in narrow channels, and other places where a great deal of engine handling was required. About one-half of the steaming has been done in the tropics with circulating water over 80 degrees F. and with correspondingly poor vacuum, so that the economy of the ship is known under all conditions. The fuel economy has proved to be excellent, being, on the average, about 25 per cent. better than the best of her sister colliers.

During the whole period of commission, two repairs have been made to the main engine. The first was to reblade the first stage of the turbine. This work was done entirely by the ship's mechanics. The first-stage blading was injured by a bolthead from the segment carrying the fixed blading. The bolt was broken off, probably through having been set up too hard while assembling. These bolts were all tap bolts and have been replaced by filister head screws with countersunk heads, and a repetition of the accident would not be possible. The accident, however, was in no way peculiar to the electric drive, as it might very well have happened to any other engine. The other repair was to replace one of the porcelain cylinders which carry the resistance; this cylinder was cracked when the ship went into dry-dock, but it was never discovered whether the cracking was due to the oiler putting undue pressure on the cylinder or due to some strain brought on it when the ship settled on the keel blocks. At any rate, the damage was repaired by two machinists in about two hours.

The amount of work expended in keeping the equipment in condition has been very small. After coming in from a run the turbine clearances are taken to see if they have changed; and the motor clearances are also checked; there has been no change in either of these clearances up to date. Before getting under way the holding-down bolts of the turbine are gone over to see if they are tight, the slip rings on the motor of the generator are examined to see if they are clean, and all insulators are also examined for cleanliness, all connections are generally gone over to make sure they are tight, the oil-switch boxes are examined to see if they have

the proper amount of oil, and the governor control-valve springs are tested to see if they have the proper tension. After starting the turbine, the emergency trip is tested. In port the oil pump is run once a week to force oil through the turbine bearings and the motor shafts are jacked daily. Due to the fact that the main condenser is used a great deal (the auxiliary condenser being too small to handle the coaling winches), the small pump for draining the turbine casing is run for a few minutes each day when in port to make sure that no water may be allowed to accumulate in the turbine.

In operation the *Jupiter's* engines have been highly satisfactory. There has been only one time when anything has happened during the handling of the engines and that was the tripping out of an exciter. The lighting set was immediately put on for exciting and the engines were ready to use again in about one and one-half minutes. If such an accident were to happen again it would take even less time, as the men are more familiar with the installation. At the time the accident occurred the ship had just left the navy yard for the first time.

The handling of the engines has been proved more than once; the ship has been handled a great deal in narrow waters and around docks. She steers very badly at times, and the quick response of the engines has more than once helped in getting out of difficulties. The engines have been used to swing ship for compass deviation without putting any way on the ship at all, the total space used for swinging being little more than the ship's length.

The ability of the turbine to stand severe abuse from water has been demonstrated several times. There is no separator on the main steam line, and several times the boilers have primed and carried considerable water over into the turbine. This was particularly noticeable during dock trials, when it was impossible to properly handle the boilers. The only indication the turbine gave was an increase in the first-stage pressure and a rattle in the casing as the water was hurled through at a high rate of speed, but the turbine blading showed no signs of bad effects from this.

The backing qualities of the engines have proved to be all that could be desired; if the ship is cruising with the resistances in, the time taken for reversing is practically nothing at all; if the resistances are out, it takes a few seconds—not more than three.

When underway the engine room is very cool, due to the fact that the generator is fitted with air impellers at each end which take their suction from the engine room, thus insuring a good circulation of air.

After all, the greatest test of the satisfactory working of any machinery is whether the men who are actually handling it and caring for it are pleased with it. If this test applies to the *Jupiter's* machinery it certainly is an unqualified success. In particular is this true if the matter is referred to the coal passers in the fireroom who have to handle much less coal than do the men on sister ships. The ship can make her contract speed of 14 knots without using forced draft at all.—"International Marine Engineering."

GEARED TURBINES FOR SHIP PROPULSION.

In the course of their paper entitled "Geared Turbines for Ship Propulsion" which was read before a recent meeting of the Institution of Engineers and Shipbuilders in Scotland, the authors, Messrs. W. D. McLaren and G. M. Welsh, devoted some attention to the subject of the gears forming part of the installation of turbine propulsive machinery on board ship. They pointed out that very little experience has yet accumulated to enable a decision to be derived at for selecting the maximum

permissible load between the wheel teeth. The teeth being of involute form have a curvature of face dependent on the diameter of the base circle and hence the teeth on a small pinion offer a less extensive bearing surface than those on a large one. Of course, theoretically, the teeth have only one point of contact, but in actual work, however, they do not come into metallic contact, having a thin film of oil between them. Hence the flatter the curvature of the teeth the larger the supporting oil pad. There can be very little doubt regarding the retention of this film of oil while the teeth engage, when it is remembered that the period of engagement is usually less than one-thousandth of a second. Another argument for the adoption of a greater intensity of pressure between the teeth for large pinions is that the effective length of the oil contact film is longer than for small pinions. In their conclusions the authors point out that in all the cases which they have considered of steamers equipped with geared turbines an advantage in coal consumption was shown at full power over the reciprocating or direct turbine type of machinery.—"Shipbuilding and Shipping Record."

FIRST STUMPF UNA-FLOW ENGINE BUILT IN AMERICA.

The development of the una-flow engine promises much for the future of the steam prime mover. It is therefore of interest to note the introduction of this engine into America. The Ames Iron Works have recently constructed the first una-flow engine, to be made in this country with the approval of Prof. Stumpf.

As is commonly known, the una-flow engine has effected an increased economy because of reduced cylinder condensation losses. This is due to the fact that the steam is exhausted at the other end of the stroke from that at which admission occurs. Hence the steam in entering is not passed over the cooler end of the cylinder, lowered, at the end of the stroke, to the temperature of the low-pressure steam. By eliminating the loss between cylinders of a compound or triple-expansion engine, the same power may be developed with a una-flow engine having cylinder dimensions materially less than those of the low-pressure cylinder of a multiple-expansion engine. The Stumpf engine built by the Ames Company shows a reduction of 20 per cent. in this respect.

The results of tests on this engine recently published, show very significant results. The engine is rated at 100 kw.; its dimensions are 15 x 16 inches, and its speed is 250 r.p.m. The best economies are 12.5 pounds of steam per I.H.P. hour condensing, and 16.8 pounds non-condensing (superheated steam in both cases). These economies are indeed remarkable for an engine of such small capacity, but the most significant feature of the performance lies in the flatness of the water-rate curves. In varying the load from almost zero to 150 per cent. of rating the greatest variation is about 2.5 pounds per I.H.P. hour. When the water-rate curves of the best multiple-expansion engines are considered, these results appear to be almost revolutionary.—"Sibley Journal of Engineering."

THE JUNKERS OIL ENGINE.

By Dr. F. E. Junge.

In an age of intensive cultivation, when efficiency is the controlling factor of human affairs, and the observation of its rules secures to all, who want to profit, increasing mastery over the means of life, it is natural and necessary that efforts of advance, as in other fields, would also be made in the realm of power generation, which is the foundation of the indus-

trial arts. Efforts certainly have been made and advances effected, some in the traditional modes of steam power generation, and some in the novel field of internal combustion.

While for purposes of large-scale power production and transmission, the steam turbine, owing to its inherent advantages, has attained and will always retain its distinct usefulness, especially when coal is the impelling fuel and stationary plants are concerned, a different outlook presents itself in those fields of application where economy, capacity and weight are limiting conditions, and where, in addition to the nature of the plant, the nature of the fuel—heat density, value, storage, etc.—must be considered.

To the qualities of liquid fuel, offering numerous advantages over solid combustibles the rapid development of oil power during the last decade must be attributed. Owing to the fact that the fuel is liquid and can be evaporated, atomized or sprayed, a very intimate mixture of combustible matter and air before and during combustion results. This, in turn, enables those operations for which the steam prime mover requires a boiler, to be performed and terminated in the power cylinder itself. Not by indirect contact—heating and expansion—but by direct contact of internal combustion is the heat energy of the fuel transformed into work. As heat influx and efflux are confined to the working cylinder—a comparatively small space—temperatures are increased tenfold, while heat losses are diminished, superior fuel consumption, economy and sootless combustion being the result.

The first stage of oil-engine development is characterized by the practice of adopting methods and using elements of machine design which had proved successful in the generation of gas power, without attempting to adjust them to the varied requirements of liquid fuels. The Otto engine and the four-cycle mode of operation were the Alpha and Omega of the earlier art. Nothing but conventional forms and ideas were employed. The results accordingly were disappointing, not only in regard to mechanical operation, but also with reference to the economic results, because comparisons of thermal efficiency, etc., were restricted to the steam and oil engines, respectively, and did not include the boiler plant and its auxiliaries. The greatest drawback of earlier development was that only high-grade oils could be used in the heat engine, while medium grades, even petroleum, gave endless trouble. Crude oils, of which we have large quantities all over the world, and residue oils remaining available as by-products of distillation, could not be used at all.

The second step in the evolution of oil power, especially in the direction of an enlarged range of application, was made by the invention of the Diesel process. It is characterized by the compression of air in the power cylinder to such temperatures that the fuel will ignite when injected into it in a finely atomized state. Though the application of the Diesel process in practice was considerably retarded by difficulties of mechanical character, which are not wholly overcome even today, it has brought about results, thermal as well as economic, which were previously deemed impossible. It has opened new avenues of thought and has called into being superior rules of workmanship which, though conventional in form, were revolutionary in effect. But the Diesel engine, pushed after the expiration of the Diesel patents by manufacturers all over the world, had soon reached a state of development beyond which further progress becomes impossible, owing to fundamental limitations which are associated with the application of the (new) Diesel process in the (old) familiar form of gas engine of the Otto type. Diesel did not recognize the mechanical limitations of existing heat engines and their bearing on the essential features of internal combustion. Nor did he conceive the idea of evolving new forms or elements of machinery, which would permit him to attain all the possibilities inherent in his process. Capacity, maneuvering, economy—in short, the essential conditions of efficient locomotion—had

reached their highest mark in the conventional Diesel engine. No further progress was possible unless the accepted principles of engine design were discarded.

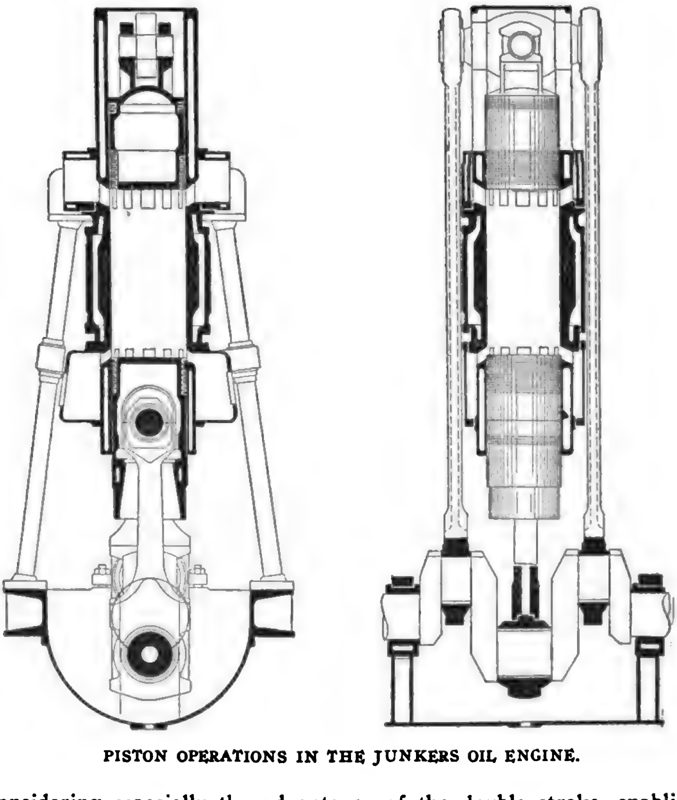
A new step in advance in the generation of oil power was made by the discovery and application of the Junkers process of perfect heat flow. Contrary to established practice, it effects compression heating, and expansion of the charge in two directions, opposite to each other, realizing high initial and low end pressures, whereby the mechanical and thermal deficiencies of the Diesel process are eliminated and its economic efficiency is increased. Prof. Junkers, known by the calorimeter and other devices which he invented, was the first scientist to realize that the limiting factors of oil-engine development were pressures and temperatures at the beginning of the cycle; he saw that further progress was impossible unless these factors were clearly defined; and he concluded that the only way to increase capacity and economy and to decrease weight and space of oil engines, consisted in abandoning the traditional engine design and turning to the employment of opposed pistons. When combined with the Junkers process of perfect heat flow, they permit the reduction of heat losses at the time when these losses are most harmful to the efficiency of operation of the engine, beside giving a perfect mechanical solution of the problem. Hence the Junkers engine permits the attainment of pressures and temperatures in the combustion space of the heat engine hitherto impossible with ordinary appliances. It raises the cycle of operation to a higher level, for example, of pressures from 100 to 150 and even 200 pounds per square inch, doing away with the limitations of intensive operation and preserving its possibilities to the fullest extent.

The practical results achieved with Junkers oil engines confirm the theoretical anticipations of the inventor in every respect. They reveal not only an increase of thermal efficiency, mechanical excellence, maneuvering quality, and ratio of capacity to compactness, but they open up new vistas in regard to the utilization of inferior fuel oils, which are of the utmost commercial consequence, especially in the western sections of the United States.

As will be seen from the drawings, shown on the page, the Junkers engine has two pistons working oppositely in one cylinder, simultaneously receding from and approaching each other during one revolution of the crank shaft. There are no cylinder covers as in the Diesel single-piston engine. At each end of the cylinder are situated ports, as far as possible arranged around the whole circumference. These ports are covered and exposed by the pistons in turns. Fresh scavenging air enters through the ports at one end after the products of combustion have escaped through the ports at the other end. The oil is injected between the pistons, at the time when they are near the dead-center position, and injection continues during the early part of the outward stroke. The mode of operation of the Junkers engine is as follows: when the two pistons have reached their outermost position in the cylinder, clean scavenging air delivered by air pumps enters through the ports at one end of the cylinder and drives the products of combustion in front of it out through the ports at the other end. The cylinder is now filled with clean air. On the in-stroke the pistons approach each other and compress the air between them to such a degree that the oil on being injected immediately ignites and burns, the resulting pressure driving the pistons outward until they uncover the outlet and inlet ports simultaneously, as described.

This mode of construction and operation results in the following general advantages: cylinder covers which in large oil engines give trouble on account of their complicated forms, are absent in the Junkers engine, as the cylinder is closed at both ends by the opposite pistons, which can be easily removed when necessary. The pressures arising in the power cylinder are directly transmitted to the crank shaft through the piston

rods, without the intervention of passive machine parts, so that the frame, bed plate, and main bearings are relieved of stress. The cylinders have no longitudinal stress. There are no stuffing boxes, which are unsatisfactory at high pressures and temperatures. The division of the stroke to two opposed pistons permits the adoption of a long stroke and wide range of expansion with only a low piston speed. Thus a long cylinder of comparatively small diameter is obtained, which in its turn permits of an advantageous form of combustion chamber and good scavenging. Finally, the arrangement of opposed pistons results in a good balance and consequently quiet running.



PISTON OPERATIONS IN THE JUNKERS OIL ENGINE.

Considering especially the advantages of the double stroke, enabling a more perfect ratio of compression and expansion, the main feature is the increased capacity. Given the same height of engine, the same number of revolutions and the same mean pressures, obtained from the respective indicator cards, the Junkers engine develops 75 per cent. more power than the traditional system. This fact is of the utmost importance in all cases when space and weight are limiting conditions. Modern fighting craft, for example, require the utmost concentration of power compatible with efficiency. As high as 20,000 horsepower is required in modern submarine construction, and the available room is confined to the utmost. Similar conditions obtain in the propulsion of air craft, where weight and fuel economy—beside maneuvering capacity and certainty of operation, de-

termine the usefulness of the prime mover. None but the double-piston arrangement can meet the exacting requirements of modern locomotion.

Other advantages are: (1) comparatively small area of the clearance space, preventing heat losses, permitting the attainment of high mean pressures (average 150 pounds per square inch), slow speeds, quick starting of the engine from cold, low fuel consumption per unit output, and ability to use all grades of fuel oils; (2) better scavenging, which produces high mean pressures, reduces the consumption of scavenging air, and decreases the power absorbed for pumping work; (3) perfect mixing of air and fuel in the power cylinder, giving good combustion, eliminating residues, increasing fuel economy, and extending range of application of inferior oils; (4) enlarged crank radius and elastic crank shaft; (5) reduction of losses in the moving machine parts, hence increased mechanical efficiency; (6) improved ratio of mean and maximum pressure, enabling better utilization of all parts which are designed to withstand the highest stresses occurring in the process; (7) reduction of weight and cost per unit of output.

A most important feature, especially in regard to the propulsion of vehicles and craft, consists in the ability to carry overloads of 50 per cent. and more, requiring no auxiliaries, but the throttling of the exhaust outlet. The Junkers engine has stretched the upper limit of oil-power capacity to the threefold of the values obtained by existing systems. Assuming that the greatest shaft diameter attainable is 600 millimeters, and therefore taking the limit of cylinder diameter at 900 millimeters, the following powers per shaft for six working cylinders are obtained as maximum: 4-cycle Diesel engine, 2,750 horsepower; 2-cycle Diesel engine, 4,500 horsepower; 2-cycle Junkers engine, 7,000 horsepower; 2-cycle Junkers tandem engine, 14,000 horsepower. Under conditions as they prevail in America, and especially in the western sections of the United States, the chief advantages are: simplicity of construction, absence of complicated valves, etc.,—these being only two moving parts in the power cylinder—an arrangement enabling cheap manufacture and fool-proof operation, before all rendering the engine capable of using inferior grades of oil, such as are abundant in California and Mexico. In the accompanying table the results of fuel-consumption trials with various grades of oil are presented, together with the analysis of the oils used. It is seen that the consumption of crude Californian and Mexican oils is only 0.48 pound per brake horsepower hour, and the thermal efficiency of the process 44 per cent., as much as 48 per cent. of asphaltum being contained in the fuel. The lower speed limit at which self-ignition takes place is between 30 and 60 revolutions per minute.

Table Showing Results of Fuel Consumption Trials in Junkers Engines.

Analysis of Oil.	Gas Oil.	Pacura.	California.	Mexican.	Masut Coal.	Tar.
Cal. Value B.T.U. per lb....	18,000		17,400	17,460	17,800	15,950
Carbon, per cent.	86.9		84.0	83.2	85.6	90.1
Hydrogen, per cent.	12.2		11.1	11.3	12.1	7.1
Sulphur, per cent.	0.71		0.87	2.43	0.05	0.68
Oxygen, per cent.	0.16		3.93	2.90	0.72	1.66
Incombustibles, per cent.	0.00		0.13	0.14	1.48	0.50
Specific gravity	0.85	0.90–0.98	0.86	0.94	0.90	1.07
Viscosity	1.14		6.41	4.11	19.9	1.66
Flash point, deg. Cent.	80	140	137		113	98
Asphaltum, per cent.			48.1	45.6	*	
Results						
Mean ind. pressure lbs. per sq. in.	154.4	161.7	155.8	147.0	158.6	147.0
Thermal efficiency, per cent.	48.0	46.0	44.7	44.3	46.5	45.0
Revolutions per minute	200	205	200	200	200	200
Brake and horsepower	160		151	145		144
Fuel consumption, lbs. per B.H.P. per hour	0.427*	0.440	0.480	0.480	0.305**	0.480

* Referred to fuel having a calorific value of 18,000 B.t.u. per pound.

**Referred to indicated horsepower.

While the Junkers engine was at first exclusively built for stationary power plants and for driving vessels, it is of late being adapted to the propulsion of locomotives, automobiles and air craft. The Junkers engine is now being constructed by the leading ship builders and manufacturers all over the world, as well as at the Junkers works in Aachen and Magdeburg. Among other builders are the Allegemeine Elektrizitats Gesellschaft, Germany, Doxford & Sons, England, Nobel and Nornowo, Russia, Dujardin, France, and the General Electric Company, United States.—" The Engineering Magazine."

REPORT ON TEST OF DIESEL ENGINE PLANT OF THE NATIONAL ICE AND COLD STORAGE COMPANY, SAN FRANCISCO, CAL.

By J. B. Howell, Member.

Report on test made to determine the fuel consumption of a 200-brake-horsepower Diesel engine at the plant of the National Ice and Cold Storage Company, at Union and Battery Streets, San Francisco, October 12, 1914.

This test was made to verify the rate of fuel consumption guaranteed by the engine builder.

DESCRIPTION OF PLANT.

The engine tested was a 4-cylinder, 4-cycle, vertical Diesel engine, having a rated capacity of 200 brake-horsepower at 250 r.p.m., manufactured by the Dow Pump and Diesel Engine Company of San Francisco.

The engine was equipped with an extension shaft carrying two belt wheels, by which it was belted to both an ammonia compressor of 100 tons' refrigerating capacity, and to an electrical generator of 65-kw. capacity, at 1,200 r.p.m., 250-volt direct current.

When running at full load the engine carried the ammonia compressor and the generator loads.

During the test made at reduced loads the generator was allowed to run without load.

The air compressor supplying injection and starting air is a Reavell, 3-cylinder, 3-stage compressor, driven directly from an overhung crank at one end of the engine crankshaft.

The principal dimensions of the engine are as follows:

Cylinder diameter	12.01 inches; area 113.28 square inches.
Stroke	18.10 inches.
Rated speed	250 r.p.m.
Piston speed	250 r.p.m. = 754.16 feet per minute.

At 250 r.p.m., each pound mean effective pressure on piston produced .6472 indicated horsepower in one cylinder.

Guaranteed fuel consumption by makers were made as follows:

At 50 per cent. load	.50 pound per b.h.p. hr.
At 75 per cent. load	.45 pound per b.h.p. hr.
At 100 per cent. load	.41 pound per b.h.p. hr.

The fuel specified by the builders upon which the above guarantees were made called for air oil of the following characteristics:

Gravity, not under	24	degrees Beaume.
Flashpoint, not under	180	degrees F. open cup.
Burning point, not under	235	degrees F.
Asphalt, not over	25	per cent.
Sulphur, not over	.75	per cent.
Heat value, not over	19,200	B.T.U. per pound.

Fuel used as analyzed by Smith Emery and Co., of San Francisco, as follows:

Specific Gravity (23.93 degrees Beaume)	.9095
Water, per cent	.08
Sand	none.
Flashpoint (open cup) degrees F	212
Burning point, degrees F	235
Viscosity (Tagliabue, water, 60°), degrees	81.6
Sulphur, per cent	.44
Asphalt, "E Grade," by distillation, per cent	23.2
Heat value lower (water incondensed) B.T.U	18,407
Heat value higher (water condensed) B.T.U	19,649

Value of asphalt content indefinite, as there are several grades.

CONDITIONS OF TEST.

The test runs consisted of a 6-hour run at full load, and a 2-hour run at approximately 75 per cent. load, an interval of 30 minutes elapsing between the runs.

Owing to the difficulty of accurately determining the power absorbed by the ammonia compressor, and the efficiencies of the belts, and the losses in generating electricity, it was agreed by the parties concerned to determine the indicated horsepower directly from the engine, and arrive at the brake horsepower by the mechanical efficiency, which was to be taken from reliable tests on Diesel engines.

These mechanical efficiencies are given herewith and are obtained from test data published by Mr. R. Royds, J. W. Campbell, M. C. E., and Mr. Longridge, M. Inst. C. E.; also from the Encyclopedia Britannica.

The data given herewith are obtained from the curve sheets attached hereto.

Loads, per cent.	B.H.P.	Friction H.P.	Mech. Efficiency, per cent.
60	120	48	69.6
70	140	51	72.55
80	160	54	74.5
90	180	56	76.3
100	200	60	77.7
110	220	62	78.9

During the test the loading was kept within a range which did not exceed 6 per cent. above the mean nor fall below 5 per cent. under full loading.

During the reduced-load test the range was kept within 2 per cent. of the mean.

Indicator and Cards.—Indicator diagrams were taken from each cylinder every 15 minutes, and as there were four cylinders the turning of cards was so arranged that not more than 4 minutes elapsed without a card being taken from some one of the cylinders.

Two indicators were used, and they were applied alternately to each cylinder, so that a complete set of cards was obtained for each indicator from all cylinders.

One of the indicators was made by the Lunkin Co., Ltd., of London, and one made by the American Steam Gauge and Valve Mfg. Co. Both of these indicators were calibrated at the University of California. The Lunkin indicator having a constant of 344 to multiply the average height of card to obtain the M.E.P. The American steam gauge having a constant 335 to multiply the average height of the card to obtain the M.H.P.

Measurement of Indicator Cards.—The measurement of the areas of each indicator card was done by two operators using different planimeters, and the length of each card was also separately measured by the two operators, and their results agreed.

Measurement of Fuel.—The fuel was accurately weighted on tested scales, and poured into a small tank located above the engine.

Cooling Water.—A record of the supply and outlet temperatures of the cooling water was kept but the amount used was not recorded, as the water is used over and over again. The supply temperature averaged 80 degrees F., and the outlet temperature ranged from 135 degrees to 152 degrees F.

Ammonia Compressor Data.—Indicator cards were taken from the ammonia compressors every 15 minutes, but no attempt was made to calculate the horsepower from these cards, as the results would have been inaccurate due to unknown factors.

Electrical Generator Data.—During the 6-hour full-load run an accurate record of the electrical power generated was kept but was not used, for an accurate check on the power developed due to unknown factors.

Injection Air.—The injection-air pressure during the 6-hour full-load run ranged closely around 925 pounds per square inch.

During the 2-hour run at reduced load the pressure remained steady at 860 pounds per square inch.

Exhaust Gases.—No attempt was made to analyze or take the temperature of the exhaust gases, but the clearness of the exhaust from each cylinder was observed. Only one of the cylinders showed signs of incomplete combustion.

Results.—The results are tabulated herein, on the 6-hour full-load run the American Steam Gauge Indicator showed signs of faulty action and the cards taken by this indicator were thrown out.

On the 2-hour reduced-load run a loose drive on this indicator was repaired and cards from both indicators were used.

G. K. Davol, consulting engineer, students from the University of California, and J. B. Howell, representing the engine builders, conducted the test.

ENGINE NO. 2.—FULL-LOAD TRIAL, 6-HOURS, 9 A. M. TO 3 P. M., OCT. 12, 1914.

Card Nos. Cyls. 1 & 2	Card Nos. Cyls. 3 & 4	Cyl. No. 1. Area	Cyl. No. 1. Length	Cyl. No. 2. Area	Cyl. No. 2. Length	Cyl. No. 3. Area	Cyl. No. 3. Length	Cyl. No. 4. Area	Cyl. No. 4. Length
2	1	.820	2.80	.900	2.82	.770	2.85	.950	2.82
4	3	.765	2.80	.860	2.82	.840	2.84	.935	2.84
6	5	.728	2.81	.770	2.82	.880	2.85	.895	2.81
8	7	.770	2.82	.870	2.80	.805	2.84	.870	2.82
10	9	.695	2.80	.870	2.82	.810	2.84	.800	2.83
12	11	.690	2.82	.920	2.83	.830	2.84	.860	2.80
14	13	.735	2.80	.935	2.82	.830	2.82	.795	2.82
16	15	.700	2.82	.900	2.82	.830	2.82	.820	2.82
18	17	.690	2.82	.845	2.80	.860	2.81	.840	2.80
20	19	.750	2.82	.880	2.81	.835	2.82	.805	2.80
22	21	.710	2.82	.845	2.80	.880	2.83	.850	2.82
24	23	.730	2.86	.850	2.81	.825	2.84	.800	2.82
		8.783	33.77	10.445	33.77	9.995	33.98	10.220	33.82
M.E.P., lbs.		89.467		106.390		101.184		103.95	

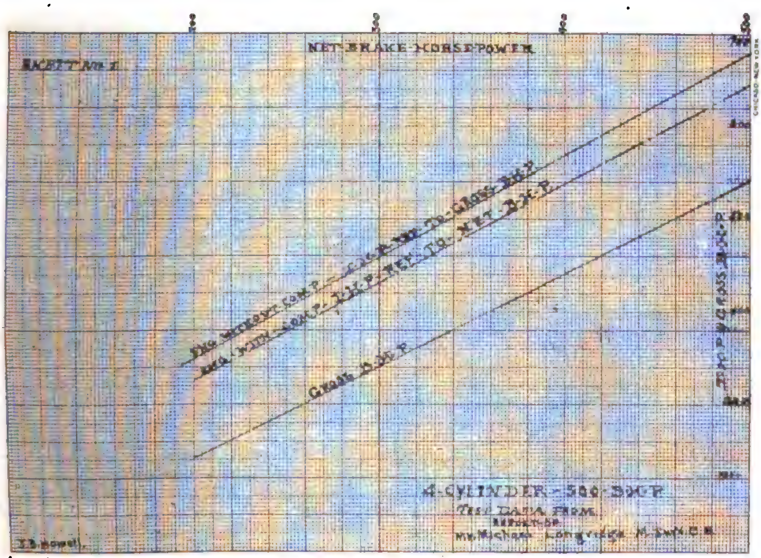
Revolutions per minute	250
Total indicated horsepower	259.52
Mechanical efficiency, assumed from curve sheet No. 304, per cent	77.3
Total brake horsepower, mean	200.93
indicated horsepower, hours	1,557.12
brake horsepower, hours	1,205.58
fuel consumed, 6 hours	485.75

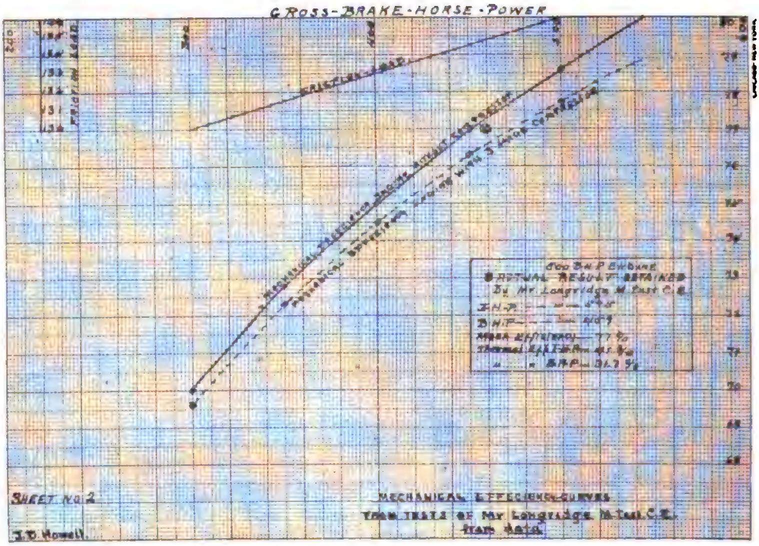
Heat value fuel per pound, B.T.U........................... 19,649
Fuel consumption per indicated horsepower hour, pound....... .3119
consumption per brake horsepower hour, pound......... .4029
Heat units supplied per I.H.P. hour.......................... 6,129.00
units supplied per B.H.P. hour.......................... 7,916.59
Thermal efficiency on basis I.H.P., per cent................... 41.5
B.H.P., per cent.................. 32.14

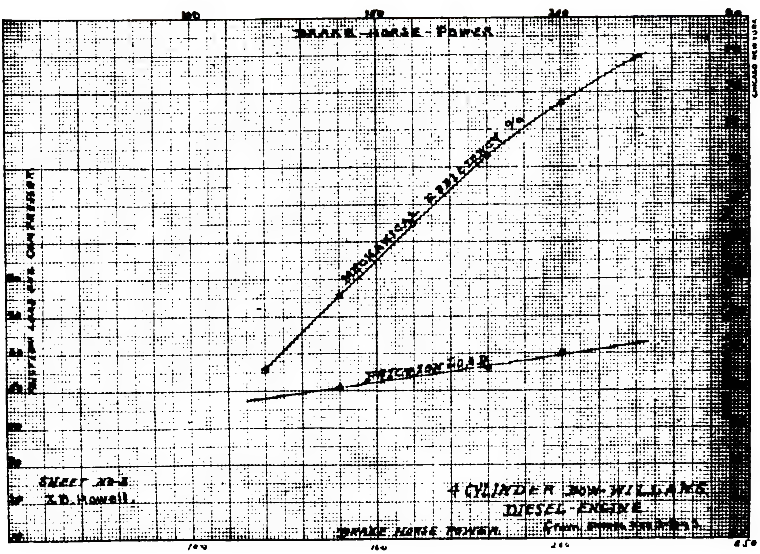
ENGINE NO. 2.—THREE-QUARTER TRIAL, 2 HOURS, 3:30 P. M. TO 5:30 P. M., OCTOBER 12. 1914.

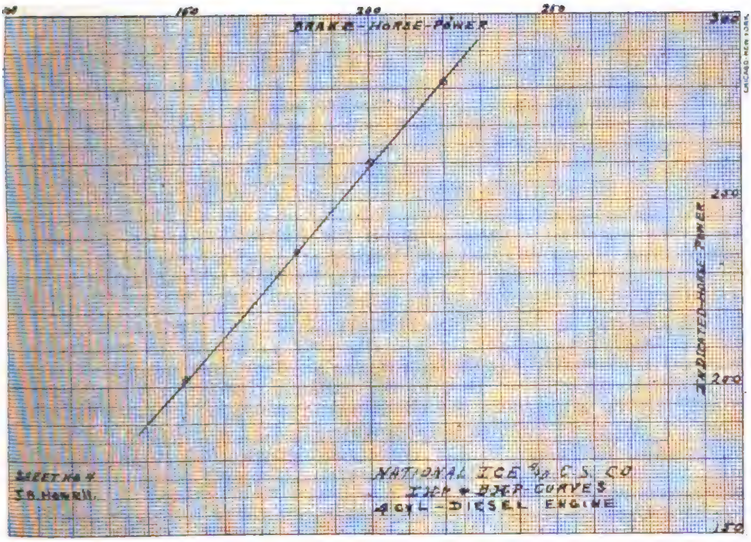
Cyls.	Cyls.	Cyl. No. 1.		Cyl. No. 2.		Cyl. No. 3.		Cyl. No. 4.	
26	25	.540	2.82	.720	2.82	.680	2.82	.635	2.82
28	27	.570	2.82	.700	2.82	.730	2.84	.660	2.82
30	29	.560	2.66	.710	2.80	.675	2.82	.660	2.82
32	31	.560	2.82	.700	2.82	.690	2.84	.655	2.82
		2.230	11.12	2.830	11.26	2.775	11.32	2.610	11.28
M.E.P., lbs.		68.985		86.457		84.328		79.594	

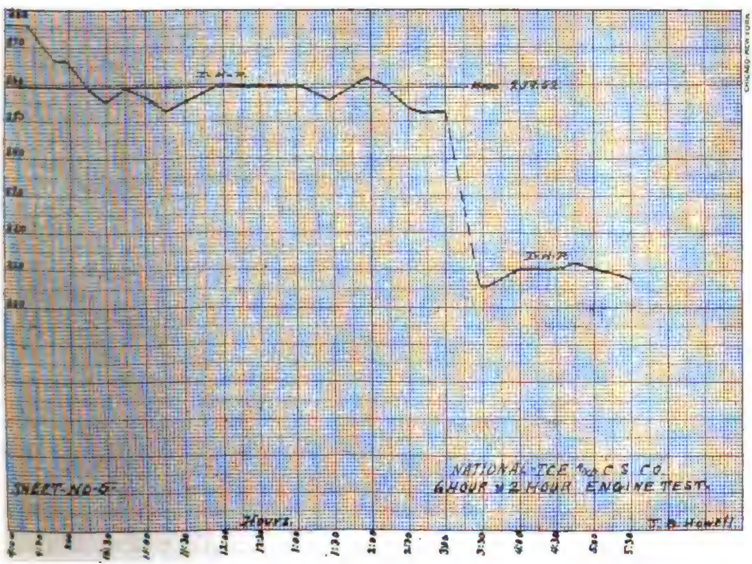
Revolutions per minute.., 250
Total indicated horsepower (mean)........................... 206.69
Mechanical efficiency assumed curve sheets, 3 and 4 per cent.... 73.9
Total brake horsepower (mean)............................. 152.74
I.H.P. hours 413.4
B.H.P. hours 305.48
fuel consumed, pounds................................ 112.5
Heat value fuel per pound, B.T.U............................ 19,649
Fuel consumption per I.H.P. hour, pound..................... .272
consumption per B.H.P. hours, pound.................... .368
Heat units supplied I.H.P. hour.............................. 5,346.5
units supplied per B.H.P. hour.......................... 7,230.8
Thermal efficiency on basis I.H.P., per cent.................... 47.6
B.H.P., per cent....................... 35.2




GROSS-BRAKE-HORSE-POWER
FRICTION LOAD
SHEET NO. 2
J. D. Howell.
MECHANICAL EFFICIENCY CURVES


BRAKE-HORSE-POWER
MECHANICAL EFFICIENCY %
FRICTION LOAD
J. D. Howell.
4 CYLINDER
DIESEL-ENGINE
BRAKE HORSE POWER.


150
200
250
BRAKE-HORSE-POWER
300
250
INDICATED-HORSE-POWER
200
150
SHEET No 4
NATIONAL ICE & C.S. CO
4 CYL - DIESEL ENGINE


SHEET-NO-5
NATIONAL ICE & C.S. CO.
6 HOUR & 2 HOUR ENGINE TEST
Hours

A HIGH-VACUUM RECIPROCATING AIR PUMP.

The production of the high vacua necessary for steam turbines requires an air pump which can withdraw from the condenser such a volume of highly aerated vapor as will enable the largest amount of tube surface to remain constantly active; but an air pump and a condenser are so interdependent that they should be regarded as one unit, as only by effective combination can the highest results be attained.

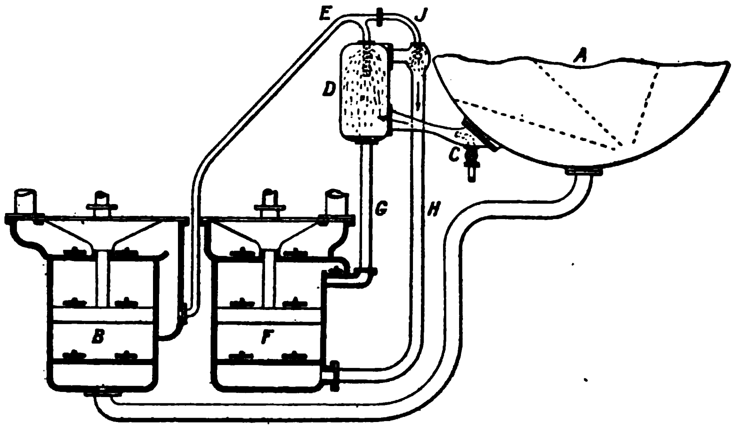
The only vacuum in a condenser which affects the economy of the turbine is that maintained in its exhaust chamber above the tubes, and as the function of an air pump is to maintain in its suction pipe the highest attainable vacuum relatively to the barometer, it follows that the resistance to vapor flow through the tube nests should be so small that approximately this latter vacuum is maintained in the exhaust chamber of the condenser. But the true resistance through the tube nests of a condenser cannot be ascertained when it is combined with a low-efficiency air pump, as under such conditions the difference between the vacua in the air-pump suction pipe and in the exhaust chamber of the condenser above the tubes may be negligible, but if the low-efficiency air pump is replaced by a steam-jet combination the vacuum in the air-pump suction pipe would undoubtedly be raised, but that in the exhaust chamber of the condenser would only be affected if the condenser were of a design which would allow of response. Failure to respond involves the triple loss of (*a*) reduced turbine efficiency, (*b*) colder condensate and therefore reduced boiler efficiency, (*c*) wasteful expenditure of pumping power in circulating the cooling water.

It has been generally accepted hitherto that the practical utility of a steam jet when combined with a reciprocating air pump lay in the maintenance of the normal vacuum when there was excessive air leakage into the system, this being secured by increasing the pressure of the jet. Whilst this is, of course, true. yet the technical value of a steam jet is at a maximum under the conditions of a practically airtight plant, because under such conditions a higher vacuum can be produced by its aid than is otherwise possible, a corollary being that the same vacuum can be produced with a smaller air pump or a smaller condenser or with less circulating water and less pumping power.

In the early application of the steam jet very little attention was given to condenser design, and because under certain conditions it was found that the jet was ineffective, its use was discredited. It has now been demonstrated, however, that provided the condenser is of suitable design, a steam jet always does increase the vacuum to an appreciable extent, and therefore always adds to the power efficiency or steam economy of any steam turbine to which it is applied.

A first essential in the use of a steam jet is that its steam is condensed amongst the feed water and the entire heat conserved; also, that the condensing is by direct contact and not by surface contact, whereby a constant efficiency of condensing is maintained. This was achieved by Mr. D. B Morison. of Hartlepool in 1907, in his kinetic-jet system, and he has now introduced the same basic idea in combining a steam jet with a reciprocating air pump. The diagrammatic sketch represents a twin air pump. The condensate is withdrawn by one barrel and discharged in the usual way. Air is withdrawn from the condenser by a steam jet which delivers into a direct-contact heater into which such a quantity of feed water from the feed tank is sprayed as will condense the steam. The air which is compressed in the heater by the steam jet into from one-third to one-half of its original volume is led to the air barrel together with a sufficiency of sealing water, the heated water from the heater passing through a non-return valve into the air barrel above the bucket. By this arrangement the technical value of the steam jet is fully realized, the

entire heat in the steam from the jet and also in the vapor withdrawn from the condenser is conserved, the size of pump for a given air duty is reduced by nearly one-half, and a minimum power is required for driving the pumps. These pumps are being used for marine geared turbines on the lines shown in the diagram, but for large land plants the condensate may be withdrawn by a small turbine-driven centrifugal, the exhaust from the turbine being utilized in the steam jet, the reciprocating air pumps being driven at a moderately high speed and at a very small expenditure of power.



HIGH VACUUM RECIPROCATING AIR PUMP.

A. Air concentrating Contraflo main condenser.
B. Barrel taking condensate from condenser.
C. Steam jet air ejector withdrawing air from condenser A and discharging into heater D.
D. Direct contact heater, into which condensate is sprayed and condenses the steam from the jet.
E. Condensate from hot well or from feed tank for condensing steam from jet in heater D.
F. Air barrel.
G. Heated water from heater D led into air barrel F between bucket and head valves.
H. Air from heater D led into air barrel F.
J. Water from hot well or from feed tank for cooling air from heater D and for sealing air barrel F.

The flexibility of the system is very great, as in the event of an abnormal leakage of air, all that is necessary in order to prevent a fall in vacuum is to supply a little more steam to the air ejector, thus preventing the mechanism from being distressed, and enabling the pump to run at such a speed as will minimize the attention required and reduce the wear and tear to almost negligible amount. As the steam for the air ejector is at low pressure, it can be taken from a closed exhaust system; in all cases, however, the steam performs the double function of withdrawing the air from the condenser, and then heating the feed water, so that neglecting radiation the thermal efficiency of the system is unity.—"The Engineer."

EVAPORATOR DESIGN.

A great deal of steam is wasted in most evaporators as fitted on board ship. As the conversion of the heat of the primary steam into vapor is a simple matter the amount of steam can easily be calculated, and if the consumption is much in excess of what it should be it will be at once indicated that something is wrong. In a single-effect evaporator the steam admitted should not greatly exceed the amount of water evaporated. When a feed heater is employed the steam used should not exceed the vapor produced by more than 10 per cent. This allows for a loss of more than 10 per cent. due to leaks, radiation and loss through the drains and for blowing down. The amount of heat lost in blowing down can be approximated by calculation. If there is 1 pound of water at 10 pounds pressure the total heat of the water above 200 degrees F. will be 987. Evaporated four times this will amount to 3,948 British thermal units. In blowing down, water at 240 degrees is replaced by water at 60 or 70 degrees. This would account for a loss of nearly 200 British thermal units if no steam was blown out, but in practice some steam is practically always blown out. There is thus an unavoidable loss of about 5 per cent. due to blowing down, which loss in actual circumstances is probably increased several times. Waste of steam can be avoided by not blowing out all the water, but little attention is, as a rule, paid to this. Evaporators are frequently in a leaky condition and possibly blowing down once in a watch is too frequent. If the saturation is taken occasionally the proper frequency can readily be determined.—"Shipbuilding and Shipping Record."

WIRELESS TRANSMISSION.

The use of sustained or undamped waves in wireless transmission has long been recognized to offer a large number of important advantages, but the generation of these waves has always been a source of difficulty. To produce an alternator of any considerable capacity which will directly generate voltages with a frequency of around 50,000 cycles is an extremely difficult proposition due to obvious mechanical and electrical difficulties.

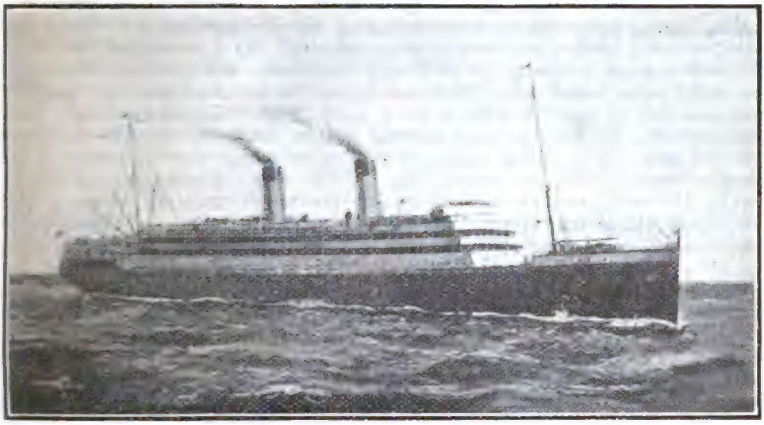
A great many of these limitations have been overcome in the Goldschmidt "reflection" alternator, such as is now operating in the transatlantic radio station at Tuckerton, N. J.

The fundamental frequency of this alternator is about 10,000 cycles, but by a ingenious system of reflection and resonance between the two windings, the output is delivered at a frequency of 40,000 cycles. The machine is of German construction, and especially rigid in design. The rotor of the alternator is driven by a 250-H.P. D. C. motor, and weighs about five tons. The speed is about 4,000 r.p.m. and the air gap is less than 1 mm. At the normal output of 110 kw., the aerial current is approximately 135 ampères, but it is claimed that the machine is capable of generating as much as 200 kw.

The receiving apparatus is ingenious in that the tone heard in the receiver is the difference tone between the transmitted frequency of 40,000 cycles and a mechanically produced frequency of about 39,500 cycles, thus giving a 500-cycle tone in the receiver. It is claimed that this system eliminates largely the interference from both static electricity and other stations.—"Sibley Journal of Engineering."

THE WIRELESS DIRECTION FINDER.

MARCONI-BELLINI-TOSI SYSTEM.



R.M.S. "ROYAL GEORGE," WHICH HAS BEEN RECENTLY EQUIPPED WITH THE MARCONI-BELLINI-TOSI WIRELESS DIRECTION FINDER.

The object of the wireless direction finder (Marconi-Bellini-Tosi system) is to enable the navigating officer of a ship to take bearings of wireless telegraph stations, with a view to finding the position of his ship or avoiding collisions with other ships. It is not claimed that bearings taken with this instrument exceed, or even equal in accuracy, those taken with an accurate optical instrument under good conditions. But it is claimed that reliable bearings may be taken with it when, owing to bad weather or other causes, direct bearings cannot be taken. Under reasonably good conditions bearings may be taken within two or three degrees, and under the worst conditions within five degrees. The accuracy of the results obtained depends almost entirely on the care with which the observations are taken, as the error due to the instrument itself does not exceed one degree. Deviation, owing to the iron work of the ship, is practically non-existent, unless the conditions are quite exceptionally unfavorable, and if it exists, is a constant factor which can be allowed for. It is not necessary to swing the ship in order to take bearings. The range of the installation is from about 5 to 50 miles or more, depending on the power of the wireless station from which signals are being received, and in the case of small ships, on the size of aerial that can be put up. It is designed to receive signals from all ordinary shore stations and ships. The direction finder requires aerial wires which are distinct from those used for the main wireless installation of the ship. The aerial system required consists essentially of two loops of equal size, suspended vertically and crossing each other at right angles. The loops ordinarily take the form of triangles of wire suspended by their apices through insulators from a triatic or other fore-and-aft stay, or from a sprit, gaff, or bracket on one of the masts. Their horizontal base wires cross the ship at the angle of 45 degrees on either side of its center line and at right angles to each other, the two bottom corners of each triangle being ordinarily made fast through insulators to stanchions at the side of the ship. Connecting wires are taken to the instruments from the centers of the horizontal base wires of the tri-

angular aerials, which are split by an insulator at their point of intersection. The range of the installation suffers to some extent if these connecting wires are very long, in addition to which the possibility of injury to the wires decreases the reliability of the installation, hence it is advisable to keep the distance between the instruments and the center of the aerial system as short as is practicable. As the readings are taken by moving an indicator to different positions and noting the points at which the sound in a telephone ceases, it is of importance that the instruments should be placed in a quiet place. The instrument gives the bearing of a wireless transmitting station with reference to the course of the ship. That is, it indicates the angle which the direction of the station makes with the center line of the ship. It shows the line on which the wireless transmitting station lies, but it does not show in what direction along that line. For instance, it may indicate a direction 20 degrees off the port bow, but it does not distinguish between this direction and that which is diametrically opposite to it, namely, 20 degrees off the starboard quarter. To use geometrical language, it shows the direction but not the sense. There will, however, seldom be any doubt as to whether the ship is approaching or receding from a land station, and, indeed, in most cases, there is only one possible way of interpreting the direction of the instrument, as by the reverse interpretation the ship would be found to be somewhere inland. If, however, there is any ambiguity, two successive bearings taken of the same station, while keeping the ship on a fixed course, will place the matter beyond doubt, and will at the same time give the ship's distance from the station by the method ordinarily in use for that purpose. In the same way the ship's position may be found by taking simultaneous bearings of two fixed stations. An obvious application of the direction finder is to find out whether the ship is on a course which will take it inside or outside a lightship or isolated lighthouse. A few signals from the lightship or lighthouse will settle the question as certainly as if the light were visible. Similarly, when making a harbor, a few signals from a station in the harbor will show immediately if the ship has drifted to one side of the entrance. When trying to locate another ship while going slow in a fog, the indication of the direction finder would show by a steadily-increasing strength of signal if the other ship were approaching, but might leave a doubt as to whether it was approaching on the port bow or overhauling on the starboard quarter; but a wireless query as to her course, addresssed to the other ship, would remove the doubt at once. The instruments consist of two separate boxes and a pair of telephones. The receiving apparatus differs very little from that in use for wireless telegraphy. A small testing instrument is provided for the purposes of adjusting the instruments and detecting any defects in the installation which might cause a wrong direction to be given. It consists of a miniature transmitter, which can be made to give signals representing those ordinarily received from outside stations. It is contained in a box on which is mounted a switch, by means of which the testing instrument can be made to give signals on either of the two waves which are ordinarily employed by ship and shore stations.

The following is a technical description of the wireless finding installation: It consists electrically of two main parts, the aerial circuits and the detecting circuits. The aerial system consists of two closed oscillatory circuits which are insulated from each other throughout and also from earth. Each of these oscillatory circuits consists of an aerial loop, in series with which are inserted a coil of wire and a condenser; the condenser being inserted in the middle of the coil of wire for symmetry. The two aerial loops, which are of equal size, are suspended in vertical planes crossing each other at right angles. The two coils of wire are also of equal size and also cross each other at right angles in vertical planes. They

are contained in a box together with their respective condensers, which are made variable for the purpose of tuning the aerial circuit to various waves. One handle varies the two condensers simultaneously. Inside the crossed coils a third coil, called the exploring coil, is mounted on a vertical spindle, by means of which it can be set at various angles with reference to the fixed coils. The detecting system consists of a pair of telephones and a crystal of carborundum, in series with a potentiometer and battery which are required to bring the carborundum into a sensitive condition. It is contained in a separate box, and is connected by wires to the exploring coil, which picks up the signals from the aerial circuits and hands them on to the detector, where they are rendered audible in the telephone. Each aerial loop is a directional aerial which receives best when its plane is in the direction of the sending station. If its plane is at right angles to the direction from which the signals are coming it receives nothing. In intermediate positions it receives signals, the induced current due to which varies as the cosine of the angle between the plane of the aerial loop and the direction of the sending station. Except in the case when one of the aerials is in a plane exactly at right angles to the direction from which signals are coming, currents are induced in both the aerials, their relative strength depending on the direction of the sending station with reference to the planes of the two aerial loops. These currents pass through the corresponding crossed coils in the direction-finding instrument, and produce in the space enclosed by them two magnetic fields at right angles to each other. The two fields, whose relative strength depends on the relative strength of the currents induced in two aerials, combine to form a resultant field at right angles to the direction from which signals are coming. The exploring coil will consequently receive the strongest signals when its plane is at right angles to that of the resultant field, or in other words, when its plane is in the direction from which signals are coming. A pointer attached to the spindle on which the exploring coil is mounted indicates the position of the latter, and consequently the direction of the sending station. In the above it has been assumed that the crossed coils are in the same planes as the crossed aerials. If the instrument is moved from this position the positions on the scale which represent the crossed aerials remain the same, so the pointer indicates correctly with reference to them.

The system is supplied by Marconi's Wireless Telegraph Company, Ltd., Marconi House, Strand, London, W.C. Its practicability and usefulness have been exhaustively demonstrated on the R.M.S. *Royal George* and *Eskimo*.—"The Steamship."

SEARCHLIGHTS.

By C. S. McDowell.

ABSTRACT OF PAPER.

Searchlights have remained practically the same for the past 25 years, although there is great necessity of an improved searchlight on account of the increased range of torpedoes and increased speed of torpedo boats.

The constituent parts of a searchlight are given in the paper and some of the essentials and desirable features of the various parts are shown. Methods of testing searchlight mirrors are given, with illustrative figures.

The results of tests conducted on Navy standard 36-inch and 60-inch searchlights and Beck 44-inch searchlights show the latter type to be much more efficient in illuminating distant objects. Relative results are shown in the figures.

Searchlights were first used during the Civil War in a very crude state; these first searchlights were fitted with metal mirrors. Later, Fresnel lenses were used to concentrate the beam. In 1876 the Mangin type of mirror was first brought out and in 1886 Schuckert, in Nuremburg, succeeded in producing a glass parabolic mirror ground to mathematical accuracy. Practically no changes have been made since that time to increase the efficiency of the searchlight; some mechanical improvements have been made, and methods of manufacture of carbons perfected, but practically the ordinary searchlight of today is the same as that of 25 years ago.

In view of the increasing speed of torpedo-boat destroyers and the increased range of torpedoes, it is very important that the searchlight, one of the principal means of defence against night attack, should be improved if possible. There has lately been developed by Mr. Heinrich Beck of Germany, a new type of searchlight much more efficient than those now in use, which is described a little later in this paper.

In considering the question of searchlights and their relative efficiencies it becomes natural to consider them as made up of the following constituent parts:

1. Searchlight, drum, pedestal, system of control in azimuth and in elevation, shutters and other purely mechanical details.
2. Rheostats.
3. Searchlight mirror.
4. Lamp mechanism.
5. Searchlight arc.

One type of searchlight may show increased efficiency over another because some of these parts have been worked out to give the very best results, but a searchlight to give the very best results must have all these details worked out separately and then joined together in the proper relation to give the maximum efficiencies.

Searchlight drum and other mechanical details are questions of design which affect the efficiency of the light very little. They should be worked out carefully, however, so that the searchlight is properly balanced, may be easily trained in azimuth and elevation from a distant control station or at the light itself. The drum should be so made that air currents cannot be set up inside, thereby causing flickering of the arc; ventilation should be sufficient so that light may be kept on at full intensity with shutter closed, or other means adopted for keeping the light burning at lower intensities with shutter closed and bringing it to full intensity instantly upon opening of shutter. It is also considered advantageous to have a permanent ammeter and voltmeter mounted on the searchlight drum, or a connection on it so that portable instruments can be connected. A ground-glass finder which shows the position of the arc, the arc length, and the variations of the positive crater from the focus of the mirror is considered an essential in a properly designed searchlight. The mirror should be so secured that it may be readily removed and the mirror and front door strips should be mounted in such a manner as to eliminate shock and breakage.

The rheostat used in connection with searchlights consists of two parts: a fixed and a variable resistance. A certain amount of fixed resistance is necessary to overcome the natural unstability of a carbon arc, and its value must be at least sufficient to give a voltage drop in the resistance equal to one-half the arc stream voltage. The variable resistance is necessary in order to obtain the proper arc voltages for various carbons and various lengths of line. The resistance elements must be of sufficient area to carry the maximum currents used without undue heating of any parts and of such material that the resistance does not change excessively with temperature changes. The steps should be not greater than one volt each.

The efficiency of a searchlight mirror depends upon its trueness to parabolic form, its trueness of surface grinding, the color and structure of the

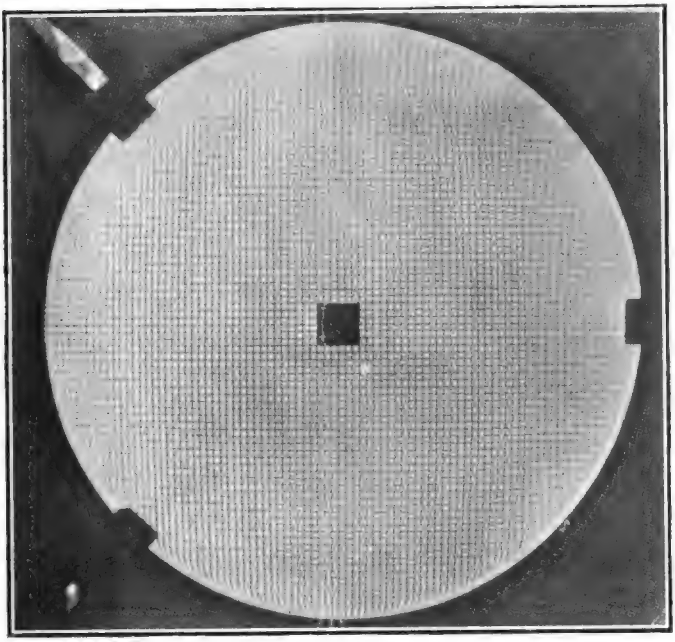


FIG. I [MCDOWELL]

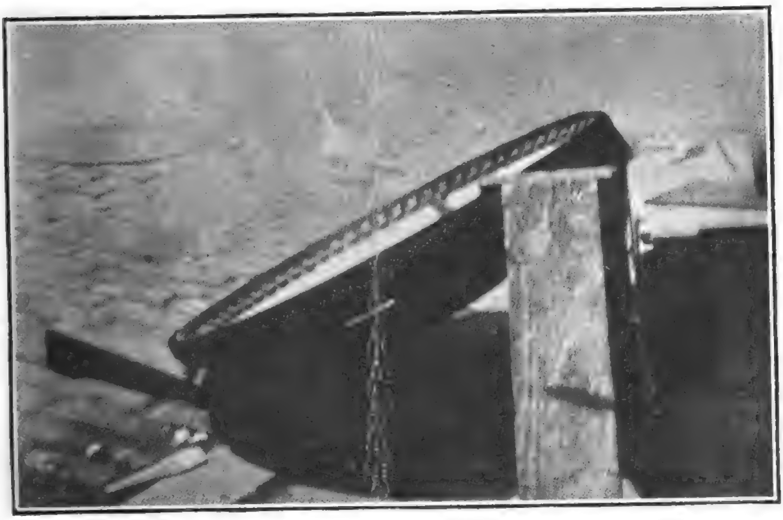


FIG. 2 [MCDOWELL]

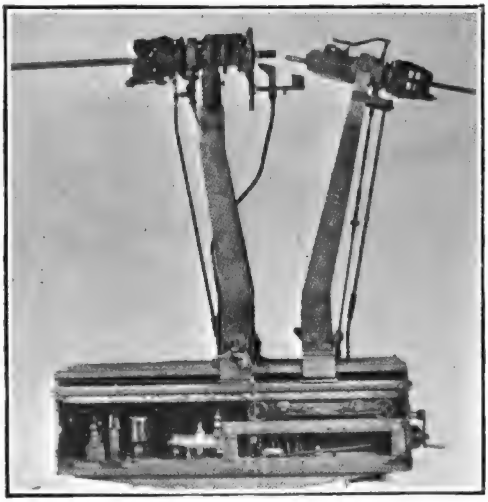


FIG. 4 [MCDOWELL]

FIG. 7 [MCDOWELL]

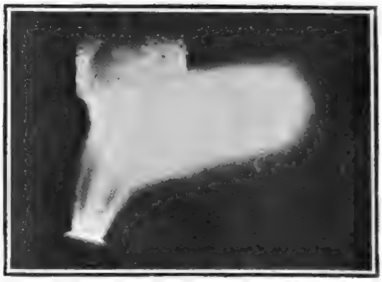


FIG. 8 [MCDOWELL]

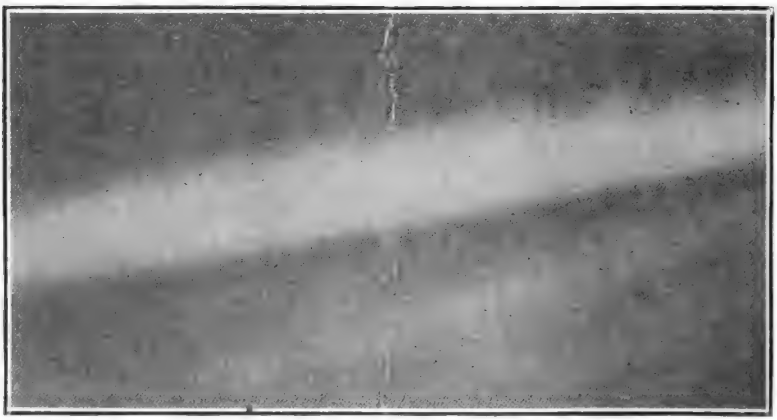
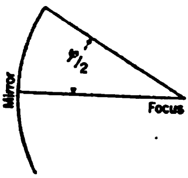


FIG. 9 [MCDOWELL]

glass, and the thickness of the glass. The losses in searchlight mirrors are due to stray rays outside of the conical beam and to absorption in the mirror. The intensity of illumination on a distant object, provided equal amounts of light fall on the mirror, is dependent on the efficiency of the mirror and the angle of dispersion (that is, at a distance, the area of the illuminated plane normal to beam).

The relation between the focal length and diameter of the mirror should be such that the effective angle φ should include the majority of light given off by the arc. By increasing the focal length, the diameter of mirror remaining constant, the ratio of the light falling on the mirror to the total given out by the arc is decreased but also the angle of dispersion is decreased, therefore, the correct focal length is that which, changed in either direction, decreases the foot-candle illumination on a distant object. If we increase the diameter of the mirror at the same time as the focal length is increased, the angle φ can be maintained constant and at the same time the dispersion decreased, thus giving greater intensity of illumination on a distant object. On board ship, however, the size of the searchlight, and thus the diameter of the mirror, is limited. A greater focal length is also of advantage in that it takes the arc farther from the mirror and thus decreases the chances of breakage of the mirror due to the heat of the arc.



In determining the efficiencies of searchlight mirrors, various tests are carried out:

(*a*) *The Line or Screen Test.*—If we imagine a section taken through the center of the mirror, a curve is obtained whose curvature should vary in a regular manner from the center towards the edge, but the curve will take a wavy form instead if the surface has not been accurately ground.

The mirror is erected opposite a screen on which are painted two systems of parallel straight lines at right angles to one another. The image of these lines is photographed by a camera placed back of the hole in the center of the screen. If both surfaces of the mirror are properly ground the lines in the photograph should be regular but not necessarily straight nor parallel. This test does not show if the mirror is parabolic or not.

An illustration of a mirror showing this test is given in Fig. 1.

(*b*) *The Sun or Zone Test.*—A perfect mirror should reflect rays from a point of light situated at its focus, in parallel straight lines. It must, therefore, also have the property of bringing a parallel beam of light, as from the sun, to a focus in a single point. This as made, consists in placing the mirrors in a plane perpendicular to the sun's rays and photographing the reflected focal point, brought out in contrast by blowing smoke on it.

An illustration of this test is shown in Fig. 2.

(*c*) *Beam Photometric Tests.*—It is considered that the most reliable test of mirrors is to actually mount them in a searchlight, place a constant source of light in the focus, and at a standard distance measure the actual foot-candles of illumination throughout the entire beam. The focal point of a mirror to be tested is first accurately obtained (our method being to place two mirrors exactly parallel with each other about 30 feet apart and place a concentrated filament lamp approximately at the focal point of the second and measure accurately the focal length of the first); the mirror under test is then mounted in a searchlight frame held in a fixed position, a concentrated filament lamp placed accurately in the focus (a 50-watt lamp is used) and at a standard distance measure the foot-candles by a portable photometer. These foot-candle readings are usually obtained every four inches across the beam on two lines perpendicular to each other and passing through the center. For the larger size mirrors

200 feet has been taken as the standard distance. Curves obtained in this test are shown in Fig. 3.

By calculating the amount of light flux falling on the mirror and that received on the distant object the losses in the mirror may be found and the coefficient of reflection obtained.

In a number of cases mirrors which showed nearly perfect grinding and sharp focal point, as determined by the first two tests, were found on this third test to reflect less light than other mirrors which had shown much poorer results on the screen and zone tests.

The lamp mechanism of a searchlight should be such as to keep the crater of the positive carbon at the focus of the mirror, should keep the arc length that which is desired, should carry the current of the carbons, should contain mechanism for rotating at least the positive carbons, should be sufficiently rigid to keep the carbons in proper alignment, and should require little care and adjustment.

An illustration of the Beck lamp mechanism is shown in Fig. 4. Both carbons are rotated for the purpose of keeping surfaces bathed in the inert gas and to keep the arc central and thus promote evenness of burning. The positive holder is fixed, the carbon being fed through it at such a rate that the crater is always maintained at the focus of the mirror; the rate of feed is controlled automatically, a small mirror being placed

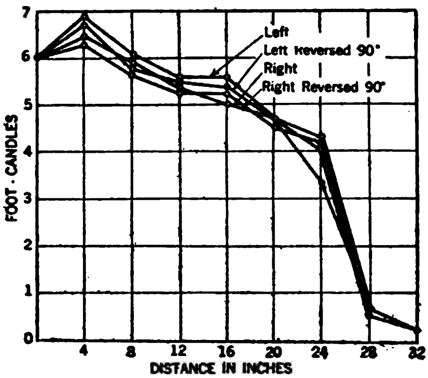


FIG. 3.—CONCENTRATED FILAMENT LAMP TEST OF THE NAVY STANDARD 36-INCH SEARCHLIGHT MIRROR TAKEN IN FOOT-CANDLES AT A DISTANCE OF 200 FEET—50-WATT, 6-VOLT, CONCENTRATED FILAMENT LAMP USED.

in the drum which, when the carbon feeds too slowly, reflects a small beam on a thermocouple, and closes a relay circuit which by means of a solenoid quickens the feed. When the carbon is back in focus the small beam of light is off the thermocouple and the feed is slowed down. In addition, the feed may be controlled by hand. The negative carbon feeds through the negative holder in a similar manner, except that the control is by hand. The negative holder is also fixed except when striking the arc, when it is moved up by a striking motor.

During tests recently conducted on searchlights the Beck lamp mechanism functioned very satisfactorily; variation of the crater of the positive carbon from the focus was about 1 mm. (hardly noticeable) and the arc length was kept practically constant. No trouble in maintaining the arc was experienced while rotating both carbons, the crater of the positive carbon remained even and there was no noticeable hissing and jumping of the arc. On the other hand, while the standard motor-controlled lamp

mechanism functioned successfully, maintaining practically a constant voltage, the arc length did not remain constant and the positive crater did not remain at the focus, due to the impracticability of constructing positive and negative carbons which will be consumed at a certain given ratio. Also the positive crater tended to wander over the surface of the positive carbon with the result that one side would project and the other recede, causing a hissing arc and making it necessary, in order to obtain any consistent results, to shut off the light and turn the positive carbon by hand about every 15 or 20 minutes.

We now come to the arc, probably the most important part of the searchlight and also the one least understood. There have been numerous treatises written on the arc, but it has been always discussed from the general illuminating side and without much consideration of its uses for searchlights. Paragraphs have been written in some illuminating treatises on the searchlight arc, but it has in general been given scant attention. This is probably more or less natural, for the authors of these books were not especially interested in searchlights, and, as a rule, had had very little experience with the subject.

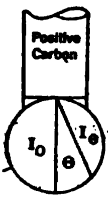
The desired searchlight arc should excel in the following particulars:

1. Small positive carbons with high current densities and thus high crater temperature throughout crater area, which gives high intrinsic brilliancy. Intrinsic brilliancy is the luminous intensity per unit area.
2. Small negative carbons.
3. Long arc length, that is, distance between positive crater and negative point.
4. Uniform mixture of carbon so as to help the evenness of burning.

The Beck searchlight may be considered as being especially designed to meet the above requirements, and therein lies its marked superiority over the searchlights now in use.

In a searchlight the angle of dispersion is directly dependent on the diameter of the source of illumination, provided the focal length is constant, and if the diameter of the source can be decreased one-half while the candle power remains constant, the intensity of light on the target would be quadrupled. Actually in the Beck light, the positive carbon is reduced one-half and at the same time the candle-power is increased so that greater efficiencies are obtained.

To illustrate the effect of arc length on candlepower it may be stated that with a circular plane radiator (the crater of the positive carbon approximates such a radiator) the maximum candlepower is obtained on the vertical axis of the radiator as shown by the sketch. Here we have I_0 = maximum intensity, and $I_\theta = I_0 \cos\theta$. Thus the smaller the angle the greater the maximum intensity of light falling on the mirror. With positive and negative carbons of a fixed diameter the angle θ depends upon the arc length, being the angle of partial shadow on the mirror projected by the negative carbon cutting off through this angle the light rays emitted by the positive carbon. This may be more clearly expressed by the formulas



$I_\theta = I_0\, q \tan\theta$ where I_0 = the normal intensity as before.
I_θ = the intensity at any angle θ from the normal, and

$q = \dfrac{\pi r^2 - S}{\pi r^2}$, where r = radius of positive carbon, and

S = the area of overlap of the negative on the positive for the angle θ.

The light intensity is reduced by the shadow of the negative carbon only for those angles of θ which are smaller than $\tan\theta = \dfrac{r + r_1}{l}$, where

r = radius of positive carbon,
r_1 = radius of negative carbon,
l = arc length.

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Thus it is seen that to decrease the angle of shadow it is necessary to increase arc length or decrease the diameter of negative carbon. The arc length is restricted to the stability point of burning.

In Fig. 5 is shown by curve the increase of candle power with increase of arc length; the candle powers are taken at 40 degrees from the normal to arc surface and are approximately the maximum.

The arc length of the Beck lamp is maintained constant at about 7/8 inch as compared to about 5/8 inch obtained at 60 volts in the standard 36-inch lamp.

Fig. 6 shows the variation of arc voltage with arc length as determined on a set of ordinary 36-inch carbons with constant current of 110 ampères.

It is, of course, well known that carbon is the most refractory of all known materials, boiling at about 4,000 degrees C., but it unfortunately commences to evaporate at a much lower temperature (about 1,800 degrees C.), so that in an ordinary arc very little of the total area of the end of the positive carbon is at the melting temperature, a small wandering spot being the real efficient part of the carbon, and the rest of the end of the carbon is consumed at a much lower temperature, giving off less intense rays and a longer wave length. This may be readily seen in comparison of the Beck and ordinary arc by the color of the arc.

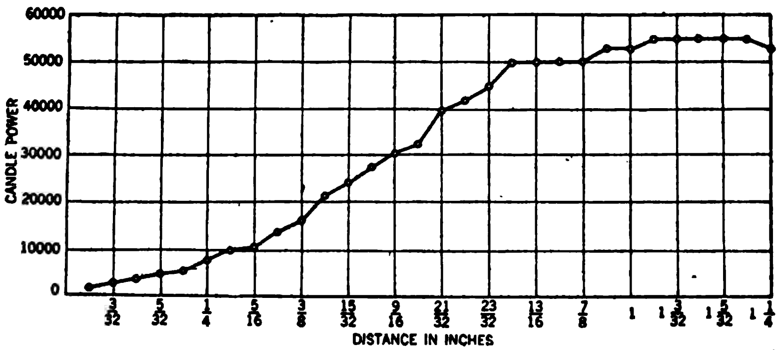


Fig. 5.—Candle Power-Arc Length Curve Taken at a Point 40 Degrees from Normal to the Arc Surface—Navy Standard Carbons—Current Normal 110 Amperes.

In the Beck arc the ends of the positive and negative carbons are enveloped in the hydrocarbon vapors which prevents the consumption of the carbons at a lower temperature, by keeping oxygen from them, in addition it cools the outer shell of the carbons, the gas being at a temperature of about 1,000 degrees C., and thus concentrates the current in the center of the carbons; thus a current density greater than 0.75 ampères per sq. mm. is obtained, and the total crater of the positive carbon reaches a very high temperature. The current is brought to both carbons near the ends by roller contacts, so the only part having this high current density is the part protected by the indifferent gas. The positive carbon is cored with a rare earth with a melting point at about 3,500 degrees C. The positive develops a deep crater, about 12 mm. deep, filled with incandescent gas. The sides of this crater reflect the light radiation to the focus of this crater, and in addition, the light from the negative is reflected, so it is believed nearly true black body radiation is obtained; and by adjusting the focus of the crater to focal point of the mirror the high peak in the luminosity curve of the beam is thus accounted for.

The area of the Beck positive carbon is 201 sq. mm., the area of the ordinary 36-inch light positive carbon is 805 sq. mm. In the Beck light the maximum intrinsic brilliancy is greater than 438 c.p. per sq. mm. The intrinsic brilliancy of an ordinary carbon arc varies from 120 c.p. per sq. mm. to 160 c.p. per sq. mm.

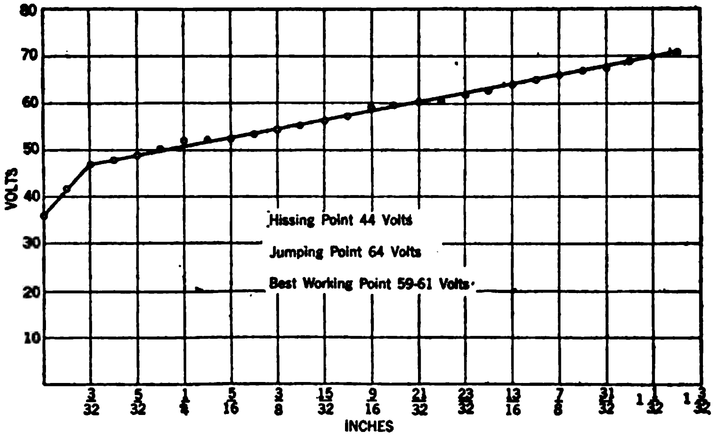


FIG. 6.—ARC VOLTAGE–ARC LENGTH CURVE—NAVY STANDARD CARBONS USED —LINE CURRENT CONSTANT AT 110 AMPERES.

In addition to the black body radiation obtained from the crater there is evidently a large amount of light radiated from the incandescent gas in the crater, of a selective nature. It would probably at first be thought that this gas should follow Kirchhoff's laws and absorb the lines which they naturally radiated, giving the Fraunhofer lines seen in the sun's spectrum; but in this case the incandescent gas is at a higher temperature than the crater and the spectrum shows positive lines apparently superimposed on the regular temperature radiation.

It is noticed that taking the Beck carbons and starting at a low current density a luminous arc stream as of a typical white flame arc is obtained, the anode being convex. As the current density is increased the anode becomes concave and the anode stream disappears, apparently being confined to the crater of the positive carbon, leaving only the lower temperature and less luminous cathode stream. The temperature of the incandescent gas within the positive crater is estimated to be between 5,000 and 5,500 degrees C. Two views of the Beck carbons burning with normal current densities are shown in the illustrations, Figs. 7 and 8.

For the luminous power J_1, which reaches the eye of the observer, we have the following equation:

$$J_1 = \frac{J}{L^4}(1 - P)^{21} K.$$

J = Luminous power of the searchlight.
L = Distance of illuminated object (under the assumption that the observer is near the searchlight).
P = Absorption by the atmosphere.
K = The coefficient of reflection of the illuminated object.

Thus if we assume that a searchlight of certain illuminating capacity makes it possible to easily distinguish an object at 4,000 meters, then a searchlight of four times the illuminating capacity will carry 5,300 meters,

assuming that the absorption of the atmosphere is 10 per cent. per kilometer and that the object is equally well distinguished in each case.

It is very hard to compare two searchlights by the eye, but during the test conducted, both searchlights were lighted (Beck and Navy 36-inch) and first one and then the other was turned on the same object with the result that objects not distinguishable, except in a hazy way, with the Navy 36-inch, were plainly outlined by the Beck light. It is apparent to the eye that the Beck light is more of a bluish white light than the standard; the ordinary searchlight beam looks yellow in comparison. The aggregate quantity of blue and violet rays in the Beck beam is about 23 per cent. At low intensities of illumination the maximum sensation to the eye for the same strengths of illumination shifts toward the blue end of the spectrum, while at higher intensities of illumination the maximum sensation for same strengths is toward the yellow part of the spectrum. This shifting of the relative sensations for different intensities of illumination is a well known phenomenon called the Purkinje effect. It is thus seen that the Beck light is particularly good for picking up distant objects where the illuminative intensity would be small.

It is also a well known fact that a colored body reflects the colors from the rays falling on it which the body itself contains, and absorbs the rays which it does not contain; the Beck light being strong in the short waves of light would thus be particularly effective in picking up objects of a bluish color, such as the various classes of ships painted bluish gray.

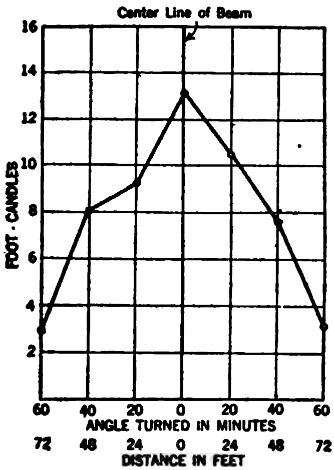


FIG. 10.—FOOT-CANDLE POWER-DISTANCE CURVE OF THE BECK 44-INCH SEARCHLIGHT TAKEN AT A DISTANCE OF 2,850 FEET —LAMP BURNING NORMALLY.

An illustration is shown, Fig. 9, of the Beck light and of a standard 36-inch searchlight, the upper beam being that of the Beck light and the lower of the standard light.

Comparative night illumination tests were conducted between standard 36-inch and 60-inch searchlights and the 44-inch Beck searchlight. The Beck searchlight beam was a more concentrated beam than that obtained from either the 36-inch or 60-inch beam of the standard lamp and the color of the Beck beam was a whiter one than that obtained from the Navy standard 36-inch and 60-inch searchlights. With the carbons burning in a normal condition and placed in the proper focal centers of their respective mirrors, foot-candle-power readings at intervals of 24 feet, were taken across the beam at a distance of 2,850 feet from the searchlights. Figs. 10 and 11 show the illumination across the beam of the Beck searchlight compared with the 60-inch standard searchlight. From these two curves it can be seen that the maximum illumination obtained from the Beck light is approximately 2½ times as great as that obtained from the standard 60-inch. In the Beck searchlight the beam shows a very high illumination at the center and falls off very rapidly at the edge of the beam. The other figures, 12 and 13, were obtained in comparing the Navy standard 36-inch searchlight with the Beck light. The figure containing the results of the 36-inch standard searchlight consists of five curves; the curves indicated by 1, 2 and 3, are plotted from actual results obtained;

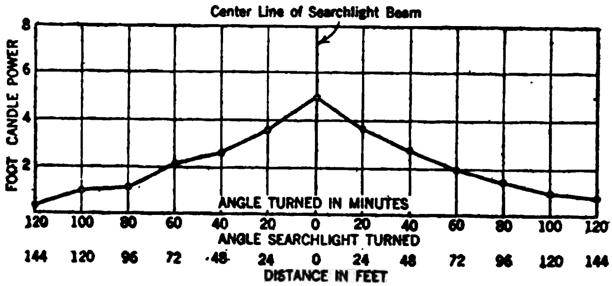


FIG. 11.—FOOT-CANDLE POWER-DISTANCE CURVE OF THE NAVY STANDARD 60-INCH SEARCHLIGHT TAKEN AT A DISTANCE OF 2,850 FEET—LAMP BURNING UNDER NORMAL CONDITIONS.

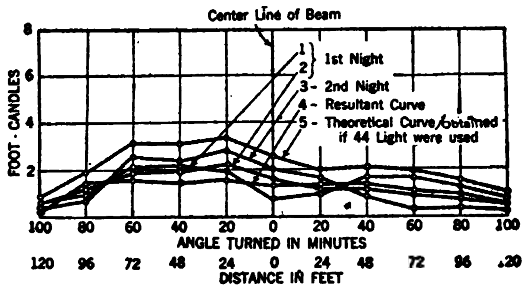


FIG. 12.—FOOT-CANDLE POWER-DISTANCE CURVE OF THE NAVY STANDARD 36-INCH SEARCHLIGHT TAKEN AT A DISTANCE OF 2,850 FEET.—LAMP BURNING UNDER NORMAL CONDITIONS.

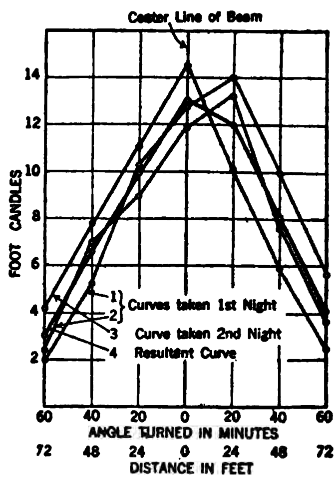


FIG. 13.—FOOT-CANDLE POWER-DISTANCE CURVE OF THE BECK 44-INCH SEARCHLIGHT TAKEN AT A DISTANCE OF 2,850 FEET—LAMP BURNING UNDER NORMAL CONDITIONS.

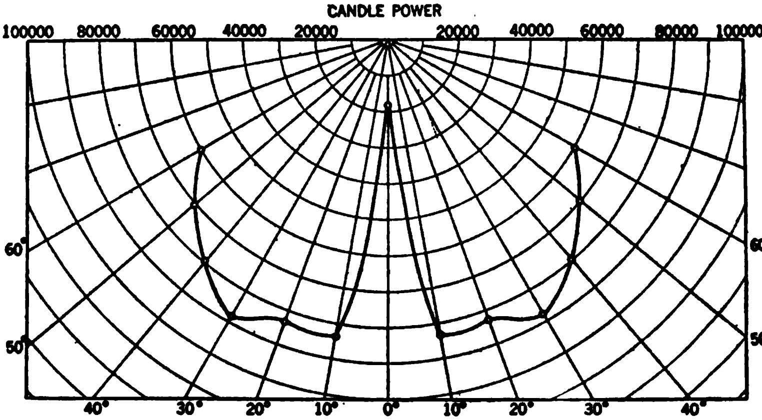


FIG. 14.—SPHERICAL CANDLE-POWER CURVE FOR THE BECK SEARCHLIGHT LAMP.

curve 4 is a resultant of curves 1, 2 and 3, and curve 5 is obtained if a 44-inch Navy standard searchlight is used in place of a 36-inch light. The other figure with the results obtained from the Beck light contains 4 curves; 3 plotted from the actual results obtained, and curve number 4 is the resultant curve of these three. From the comparative data obtained, it can be seen that the illumination obtained with the Beck searchlight is about five times as great as that obtained from the standard light. The spherical candle power of the Beck arc was measured and the results obtained are shown in Fig. 14. The maximum candle power obtained with the use of our present Navy standard carbons is 45,000 candle power, as against 88,000 candle power obtained from the Beck lamp. A zonal candle-

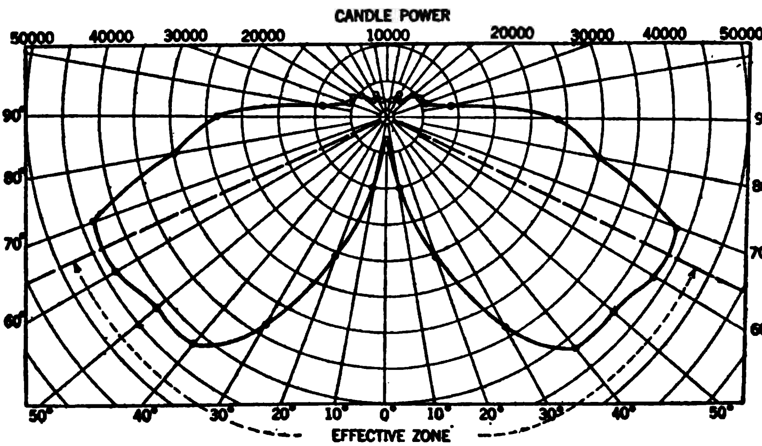


FIG. 15.—CANDLE-POWER DISTRIBUTION CURVE OF NAVY STANDARD CARBONS —LINE CURRENT 110 AMPERES—ARC VOLTAGE 60 VOLTS— ARC LENGTH CONSTANT AT 19/32 INCH.

power curve of standard 36-inch carbon arc with arc voltage at 60 volts is shown in Fig. 15.

The results shown in Figs. 10, 11 and 13, are relative only, and cannot be used to determine the zonal candle power of the beam reflected from the mirror. The absorption of the atmosphere would have to be taken into consideration to obtain the actual candle power of the beam; the results obtained in Germany with this same light were much higher than those shown here, but the relative intensities between the ordinary arc and the Beck arc were approximately the same.—"Proceedings American Institute of Electrical Engineers."

THE RESISTANCE OF THE FULL SHIP.

Experimental work on ships' models has revolutionized the forms of warships of all nations. If the performances of the modern battleships of the United States Navy be compared with the records of older ships a remarkable advance will be observed. The British Government has for many years been guided by the Haslar establishment, and the uniform excellence of the trial results of British warships bears testimony to the value of model experiments. The forms of French, Italian, Russian and German warships likewise bear the impress of experimental investigation. In the merchant service also the best is being gradually introduced, and the record-breaking propensities of the express Cunarders, the splendid performances of many cross-Channel steamers, and the economy of many of the slower cargo and passenger ships have been ensured by experimental work on ships' models. The bulk of the ships built in this and other countries are for cargo carrying only, and in this type economy in propulsion is an essential of commercial success. Experimental work has shown that great savings may be effected by a judicious selection of form. In this connection the work of Prof. Sadler of Michigan University, of Constructor D. W. Taylor of the Washington tank, and of Mr. G. S. Baker of the National Physical Laboratory, guides as to the directions in which savings may be effected.

Some very important work on full merchant-ship forms has been accomplished by Prof. Sadler, the results of which are to be found in Vols. 15, 16 and 17 of the Transactions of the American Society of Naval Architects and Marine Engineers. He has experimented with models ranging from .5 to .85 block coefficient and of normal and extreme proportions, in deep and in shallow waters. In experiments dealing with vessels of 8 beams to length and of .733 block coefficient he found that long parallel middle body and fine ends was preferable to no parallel middle body and full ends, and that the residuary resistance in the latter model was double that of the model with the fine ends at the speed proper to such a block coefficient. At this speed there was found to be a difference in total resistance of from 20 to 30 per cent. between the extreme forms. In succeeding experiments dealing with block coefficients of .653 and .855 the effect of alteration to the form of the sectional area curve was further investigated. The results of these experiments indicated that up to forms of about .8 block coefficient it is of decided advantage to have a long parallel middle body and a fine bow, but beyond that fulness round and easy bow and buttock lines are more effective in reducing resistance than having fine ends.

In Constructor D. W. Taylor's book, "The Speed and Power of Ships," the results of experiments on full ships are given amongst other valuable information. The special work to which attention is here directed is that dealing with the optimum length of parallel middle body for full

ships. As indicated in the previous note regarding Prof. Sadler's work, a diminution in resistance accompanies introduction of parallel middle body, and the results of Taylor's investigations go to show that the lengths of parallel middle body which would give the minimum of resistance are respectively 16, 27 and 35 per cent. for prismatic coefficients of .68, .74 and .80. In very full ships of just over .80 block coefficient from 40 to 50 per cent. of the vessel may be the same form as that midships.

Mr. G. S. Baker, in a paper read this year before the Institution of Naval Architects, on the resistance of merchant-ship forms, gave further light on the resistance of very full ships. The prismatic coefficients of the full models tried range from .67 to .843, which covers practically all fulnesses of cargo ships. In the fullest form tried 50 per cent. parallel middle body was adopted, which is probably just a little more than Taylor's experiments would suggest as being the optimum, but the results obtained warrant the adoption of even as much as this in the fullest vessels. One of the features of Mr. Baker's results is the influence that the run has on the resistance of full ships. A moderate fining of the prismatic curve aft brought about a reduction in resistance of 15 per cent., whilst the displacement was reduced by only 4 per cent. This finding, noticed in conjunction with a finding of Sadler that a reduction of 1.25 per cent. in displacement gave a reduction of resistance of 10 per cent., seems to indicate that it may not be advisable to go above a certain block coefficient even for slow cargo ships, as every increase in displacement over a certain coefficient means a more than proportionate increase in power. This limit seems to be reached with a block coefficient of about .78, although it is possible that considerations of effect of weather on sea performances may indicate that even that is too full for the most economical cargo ship. A deal depends on the proper distribution of displacement apart from the actual fulness of the ship, and a careful study of the work of these experimenters will repay the searcher after the form of least resistance for full cargo ships.—"Shipbuilding and Shipping Record."

LAUNCHING DATA FOR A BATTLESHIP.

By Naval Constructor John G. Tawresey, U. S. N.

ABSTRACT.

The launching of a great ship is always a matter of interest to the profession and one of anxiety to those responsible. The experience from an unbroken series of successful launches, backed by the knowledge that every condition has been investigated and every precaution taken, is not sufficient to entirely banish the thought that there may be some unusual factor for which allowance has not been made. The data for some previous vessel, especially one of corresponding dimensions and weight, are the basis of the launching arrangements adopted for a new vessel; and the great value of such data, when full and reliable, is considered sufficient reason for adding the following notes on the launching of a battleship to the valuable papers on launching already contained in the Transactions of the Society.

The launch of the *Oklahoma* was entirely successful, and the statement of data is presented as showing regular practice, not as an example of novel methods. The arrangement for distributing the pressure at the fore poppets is effective, and is presented as an alternative to the use of crushing chocks. Checking arrangements were not used. It may be worth noting that the retarding effect of rope stops, as generally used

for checking, is greater than the work done in stretching and breaking the stops.

The moving-picture method of observation was successful and is accurate. The arrangement used is fully described in the paper, also a more convenient arrangement recommended for any future observations. Attention is invited especially to the enlarged part of a curve determined photographically, and to the closeness with which the usual stop-watch observations correspond to this curve. The proof of accuracy is the agreement of the distance between stations as determined from the microscope readings from the film and the known distances apart at which they were located.

The distance the vessel ran beyond the calculated point before pivoting actually occurred is interesting. It is caused mainly by the effect of the wave on the distribution of displacement, and tends to somewhat increase the pressure on the fore poppets when the vessel lifts. The form of wave profile on the ship just before pivoting, also the very small movements just at the start of motion, are suggested as interesting parts of the launching operation for further investigation by the photographic method.

It is not suggested that moving-picture data are desirable for all launches; the curves show that stop-watch observations, carefully made, give reliable information. It is to be noted, however, that dimensions and launching weights change; practices, precautions and other launching conditions change; the kind and quality of materials that must be used change; and it is advisable that launchings be rigidly investigated from time to time and that accurate observations and records be made. Photographic methods offer advantages to that end.

LAUNCHING DATA, U. S. S. "OKLAHOMA."

Type of vessel	Battleship, First Line
Builders	New York Shipbuilding Co., Camden, N. J.
Date of launch	March 23, 1914

Principal Dimensions, Etc.

Length between perpendiculars	575 feet
Breadth on load waterline	95 feet 2½ inches
Mean draught	28 feet 6 inches
Displacement	27,500 tons
Percentage completion at time of launch	61 per cent.
Weights worked into vessel	11,765 tons
Temporary weights on board	85 tons
Men, tools, and dunnage	115 tons
Launching cradle	470 tons
Total launching weight	12,435 tons
Draught after launching, forward	13 feet 7 inches
Draught after launching, aft	16 feet 4 inches

Building Slip, Launching Ways, Cradle, Etc.

Declivity of building slip, per foot	11/16 inch
Declivity of keel blocks, per foot	11/16 inch
Length of ground ways over all	644 feet
Length of ground ways submerged	167 feet
Width of ground ways (each)	60 inches
Thickness of ground ways	16 inches
Declivity of ground ways, per foot	11/16 inch
Camber of ground ways	None

Inclination of ground ways, transverse, per foot..................3/8 inch
Spread of ground ways, center to center..................28 feet 6 inches
Material of ground ways....................Yellow pine faced with oak
Length of sliding ways over all....................................473 feet
Length of sliding ways, effective....................................465 feet
Thickness of sliding ways..15 inches
Maximum width of sliding ways................................58½ inches
Effective width of sliding ways....................................57 inches
Bearing area ....4,420 square feet
Material of sliding ways....................Yellow pine faced with oak
Length over which poppets distribute pivoting pressure............23 feet
Area under poppets to take pivoting pressure...............220 square feet
Projected length of curved bearing surface under fore poppets,
21 feet 9 inches
Projected area of trunnion segment to take pivoting pressure,
208 square feet
Number of wedges, spaced 17-inch centers..........................590
Wedges, oak, 9 feet by 6½ inches, tapered ½ inch per foot.
After end of sliding ways at start, up from water's edge............6 feet
After edge of rudder at start, up from intersection of prolonged
keel line and water, about...9 feet
After perpendicular to after poppet bearing on ways................66 feet
After perpendicular to after poppet bearing on ship................78 feet
Forward perpendicular to forward end of sliding ways.............44 feet
Forward perpendicular to center of fore poppets.................59 feet
Means for distributing pressure at fore poppets,
Rocker formed as a segment of a trunnion

Pressure, Velocity, Coefficients, Etc.

Total weight on ground ways..................................12,435 tons
Initial pressure per square foot.................................2.82 tons
Maximum pressure per square foot over ground way ends.........5.3 tons
Center of gravity forward of center between perpendiculars.......7.9 feet
Distance traveled to calculated pivoting..........................459 feet
Distance traveled to observed pivoting............................569 feet
Distance traveled from calculated pivoting to end of ways.........181 feet
Calculated buoyancy at pivoting..............................8,875 tons
Pressure on fore poppets at pivoting (neglecting buoyancy of
packing) ....3,090 tons
Pressure per square foot under fore poppets at pivoting.........14.05 tons
Pressure per square foot on projected area of trunnion segment at
pivoting ....14.85 tons
Distance traveled to bring center of gravity over end of ways.....410 feet
Depth of water over ends of ways (actual corresponding to 5 feet
10 inches tide)..9 feet 6 inches
Least excess of calculated moment to prevent tipping.......52,200 ton-feet
Static drop of vessel where poppets leave the end of ground ways,
6 feet 4½ inches
Average velocity first 100 feet (5.8 feet per second).............3.3 knots
Maximum velocity (23.6 feet per second)................Nearly 15 knots
Ship hung 2 or 3 seconds and then started slowly without application of rams or other means.
Travel before rudder entered water, about.......................25 feet
Travel to point of maximum velocity, about.....................100 feet
Time to point of maximum velocity..........................17.25 seconds
Travel to point of drop, fore poppet at end of ways, about.........640 feet
Time to point of drop, about.................................41.6 seconds
Coefficient of starting (sticking) friction, about....................0.0575

Coefficient of friction, average first 50 feet, about....................0.026
Coefficient of friction and resistance, average first 100 feet, about....0.016
Coefficient of friction and resistance, minimum, about.................0.004
Coefficient of friction and resistance, average after 250 feet run, about 0.06

Miscellaneous.

The recorded weight at time of launch were substantially confirmed by the observed displacement.

Method of release—hydraulic triggers.

Pressure used for triggers—Started with 250 pounds per square inch and was gradually increased as blocks were removed to a maximum of 1,600 pounds per square inch just before dog shores were dropped.

Temperature on day of launch on building slip, 31 degrees F.

Lubricant on ground ways—Under fore poppets applied several weeks before launching. First a thin coat of stearine, next a 9/16-inch thick coat half tallow, half stearine.

On remainder of ground ways under sliding ways, applied two to three weeks before launching. First a thin coat of stearine, next a mixture of three parts tallow to one part stearine.

Remainder of ground ways to a point 36 feet beyond low water mark, applied by means of cofferdams three to four days before the launch, and touched up the day before on account of damage from ice—Tallow mixed with lard oil.

Lubricant on sliding ways, applied two to three weeks before launching —Forward 80 feet thin coat of stearine, whole length with launching grease smeared on lightly.

Lubricant on pivoting surface under fore poppets—A mixture of half stearine and half tallow applied 9/16-inch thick.

Removing blocks, etc.—Some of the blocks at ends of vessel and most of the shores were removed the day before the launch.

Wedges were set by mauls and alternate keel blocks removed, starting at 7:30 A. M. on day of launch. Rallies on the wedges, using rams, were made at 10:00, 10:11, 10:22 and 10:32 A. M., the last block was out at 12:10 P. M., dog shores dropped at 12:13 and ship released at 12:15 P. M.

About 280 men were employed below the ship; some 40 of these afterward went on board, making about 120 men on the ship at time of launch.

Vessel was launched just as tide was turning ebb; there was a moderate breeze from the southwest.—"International Marine Engineering."

NEW DRY DOCK FOR SAN FRANCISCO.

Two dry docks are now maintained on the Pacific Coast for the vessels of the United States Navy; one at Mare Island, on a branch of San Francisco Bay, the other at Bremerton on the Puget Sound near Seattle. With the opening of the Panama Canal naval operations on the Pacific assume a much more important aspect than heretofore. Neither of the above docks will be adequate for the superdreadnaughts that will frequently need overhauling on the Coast. For this reason a contract has been executed between the United States and the Union Iron Works Dry Dock Co., of San Francisco, whereby the latter corporation will construct a $2,000,000 dry dock in San Francisco Harbor, under the guarantee that the Government will furnish at least $50,000 in business each year for six successive years.

The new dock will be constructed at Hunter's Point, a promontory extending into the bay from the east shore of San Francisco, where

two small dry docks have been in service for a number of years. The foundation will be of solid rock and the structure of reinforced concrete, specially designed to withstand severe earthquake shock. Its length over all is 1,096 feet, breadth between vertical walls 120 feet, breadth to top of altars 140 feet 8 inches and depth from mean high water to top of sill 42 feet 6 inches. Its bow is elliptical and its sides parallel. They arise vertically from the floor to within 12 feet of the top of the coping, where altars extending for almost the entire length on both sides increase the width from 120 feet to 140 feet 8 inches.

From the center to the side drainage gutters the concrete floor has a chamber of 12 inches. On it are the center keelson working platform and bilge blockways, both of wooden construction. Over the rudder pit, located near the outer end of the dock, the center keelson can be removed in sections.

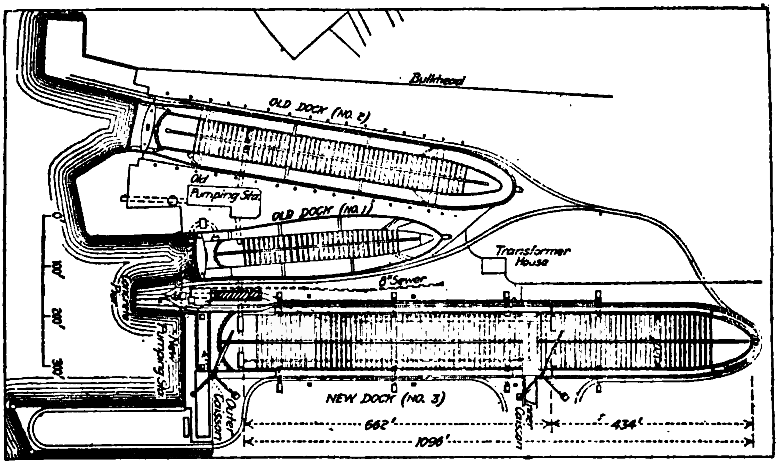


FIG. 1.—LOCATION OF TWO OLD AND ONE NEW DRY DOCKS AT SAN FRANCISCO.

The dock will be closed by an outer caisson and opened by floating and sliding this into a transverse recess in the side of the concrete structure. A similar recess and second caisson are located forward of the center, so that the dock may be used as a whole by sliding only the outer caisson into position; or the upper end alone may be used by closing the middle caisson, or both ends may be used independently at the same time by sliding closed both caissons. The center caisson recess is drained through a tunnel 4 feet 6 inches in diameter, which leads to the pump pit. While the caissons themselves are essentially of the sliding-box type, they can also be floated from one place to another and thus divide the structure in compartments of any desired length. They are designed so as to have ample stability without fixed ballast both when floating light or when partially flooded. This is accomplished by transverse partitions which divide the caissons themselves into three compartments, the middle one of which can be flooded or emptied without lowering the metacenter below the center of gravity.

Four inclined tunnels 10 feet in diameter, two on each side of the dock leading to the forward and rear sections, and two open-curved stairways at the elliptical bow, provide ready access to the interior. The

tunnels are closed by butterfly valves. To unwater the dock these are opened and the water allowed to drain to the pump pit, from which it is by-drawn by 54-inch vertical centrifugal pumps of the single-stage, volute type. Each pump has a capacity of 75,000 gallons per minute, pumping against a head up to 42 feet. They are direct connected to 750-H.P. motors, which operate at 250 r.p.m. Pump casings are of the volute form, of cast iron, in sections provided with vertical and horizontal joints, so that they may be easily removed through openings in the motor floor above. Impellers of the inclosed type will be of a bronze with a single-bottom opening, and are designed to eliminate as far as possible objectionable end thrust. The top heads are sufficiently large to permit the removal of the impellers for repairs.

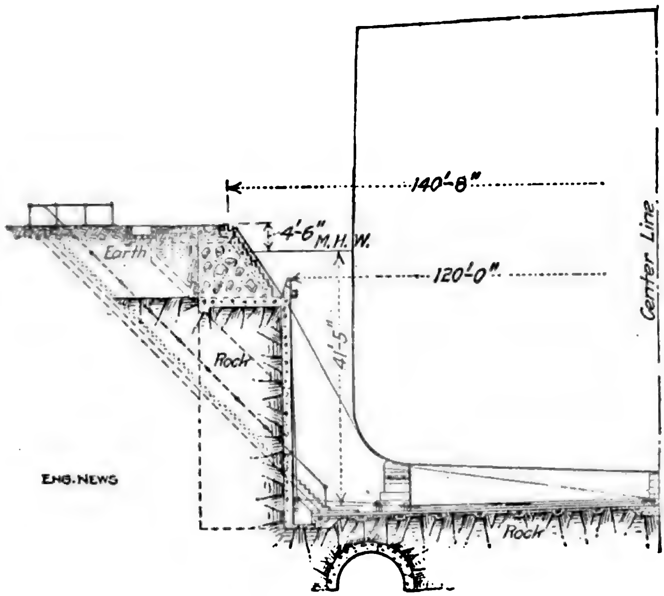


FIG. 2.—HALF-SECTION THROUGH NEW SAN FRANCISCO DRY DOCK.

Pump house and pit are to be located near the outer end of the dry dock. The top of the pit will be 48 feet below high water or 62 feet 6 inches below the top of the dock coping. Besides the four 54-inch pumps for unwatering the docks, the equipment comprises two 15-inch pumps for draining the tunnels, an 8-inch salt-water high-pressure pump, a priming pump and the operating gear for the gate valves and for the butterfly valves which close the tunnels leading to each section of the dock.

For flooding the dock a tunnel 12 feet in diameter, controlled by a butterfly valve, leads from the bay to the pump pit. Thence the water flows into the dock through the same tunnels by which the structure is unwatered. The operating gear which contains the flooding butterfly valve is housed separately from the main pumps in a small concrete building. The dock with tunnels has a capacity of 5,715,000 cubic feet and with the four large pumps can be emptied in less than two hours and twenty minutes. It can be flooded in about thirty minutes.

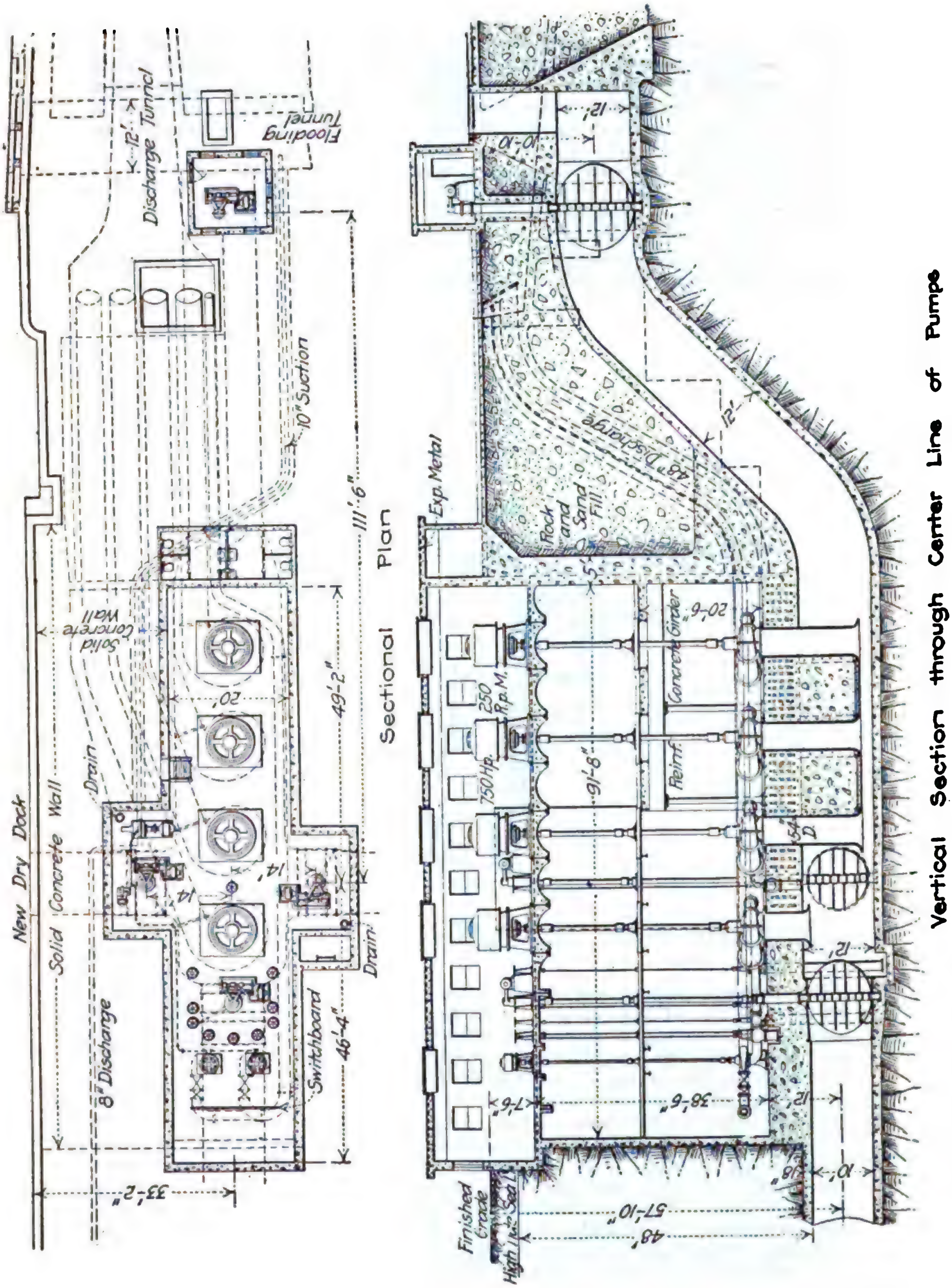


FIG. 3.—SECTION THROUGH AND PLAN OF PUMP HOUSE, SAN FRANCISCO DRY DOCK.

On the port side of the dock will be a reinforced-concrete pier, with railroad tracks so arranged that cars may be landed from barges on the bay. A reinforced-concrete wharf on the starboard side approximately 450 feet in length and 100 feet in width at the outer end, will furnish ample wharfage facilities. Electric power will be used for the pumps and machinery as well as for the compressors that will operate pneumatic implements used in ship repair. A transformer house will therefore also be required. This will be of concrete and will be erected on the port side, opposite the inner caisson recess.

Below is a tabular description of the dock and caissons:

Length from outer meeting face to head of dock, feet	1,096
Length from inner meeting face to head of dock, feet	434
Length from outer meeting face to inner meetings face, feet	662
Width of caissons, feet	22
Usable length of outer section of dock, feet	640
Usable length of inner section of dock, feet	434
Breadth between vertical walls, feet	120
Breadth at top in way of altars, feet and inches	140–08
Height of coping above higher water, feet and inches	4–06
Outer sill below high water, feet	40
Depth from top of sill to top of coping, feet and inches	44–06
Top of keelson below top of sill, feet and inches	1–05
Height of keel blocks above top of keelson, feet	4
Top of keel block above top of sill, feet and inches	2–07
Top of keel block below high water, feet and inches	37–05
Flood camber, inches	12

	Inner caisson.	Outer caisson.
Length, molded, feet	124	124
Breadth, molded, feet	22	22
Extreme height at side, feet and inches	48–06	46–06
Light draft from third deck, feet and inches	8–06	8–03
Light displacement, tons	620	620

Accessories included in the specifications are keel blocks, bilge blocks, air and salt-water piping, fresh-water hydrants, electric conduits, electric capstans, one 50-ton electric self-slewing stiff-leg traveling crane, cast-iron bitts, belaying pins, ladders, blocks, etc.

The dock was designed and specifications prepared by Hugo P. Frear, Architect for the Union Iron Works Co. and Chief Engineer for the Union Iron Works Dry Dock Company.

The new San Francisco Dry Dock is among the five new dry docks which have a total length of 1,000 feet or over, all of which can dock any vessel now afloat. Two of these docks, the Gladstone at Liverpool and the Alexandria at Bombay, are completed and those at Boston and Levis are now under construction. The general dimensions of the five are as follows:

	Length.	Breath.	Depth H.W.
Boston, Mass., feet	1,200	120	44.8
Levis, Que., feet	1,120	120	40.0
San Francisco, Calif., feet	1,096	120	42.5
Gladstone, Liverpool, feet	1,050	120	46.0
Bombay, India, feet	1,000	100	33.3

—"Engineering News."

SURPRISING FAILURE OF STEEL SHIP PLATES.

Considering the treatment which ship plates must undergo, both in the shipyard while being flanged and riveted, and afterwards in the ship in service when the plates are subjected to unknown stresses, the necessity for a method of testing the steel of which the plates are made to determine its suitability for the purpose is of the utmost importance. Lloyd's Rules for testing shipbuilding materials are generally regarded as reliable, and steel which is passed by Lloyd's surveyors is generally accepted as suitable for the purposes intended without further investigation. A case is cited, however, by Mr. W. J. B. Wilson in a paper recently read before the North East Coast Institution of Engineers and Shipbuilders, in which a consignment of steel ship plates, after passing Lloyd's tests and being characterized by the surveyor as "excellent," failed in a surprising manner shortly after being worked into the ship.

In this case the shipbuilding firm received an order for an ice-breaking passenger and cargo steamer to be built to Lloyd's highest class under special survey. The vessel was begun in November, 1907, but, owing to delay in obtaining the material, the shell plating was not begun until the end of December. B strake was first worked, but, owing to the difficulty with the flanging of the garboard strake plates, several of which cracked, the rest of the shell was practically completed before the garboard strakes were put in place.

The garboard strake plates which cracked were carefully examined, but as the results of both bending and tensile tests were excellent, the failure at the time was ascribed to the severe cold weather prevailing, the temperature being from —13 to —18.4 degrees F.

The riveting of the shell was well under way before the keel riveting was begun, but, after a few garboard plates had been riveted, cracks were noticed between the rivets and the edge of the plate in the flanged portion. These plates were condemned by the Lloyd's surveyors and were cut out. During the process two of them cracked badly in the neighborhood of the keel rivets, and the astonishing fact developed that a single blow with an ordinary hand hammer was sufficient to knock pieces out of the flanged portion.

This sudden change in these plates, which had withstood flanging, punching, riveting and, in some cases, calking, was very remarkable, and the job of riveting the keel was at once stopped until a thorough hammer test of the flanged portion of all the remaining garboard plates had been made. When this proved perfectly satisfactory, the riveting was resumed and only one or two small cracks developed.

About this time severe cold weather set in again, and on January 11, 1908, one of the garboard plates, which had been riveted in the ship for a period of three weeks, was found to be badly cracked. The crack started from a rivet hole in the flanged portion and went off into the plate in a transverse direction, being about 10 inches long and about 3/32 inch open at the widest part. This was clearly a case of spontaneous rupture.

When this plate was cut out the material in way of the keel rivets cracked in the same way as that described above, and a new hammer test was made with startlingly different results from the previous one, as every plate in the garboard strakes, with the exception of those previously replaced, cracked and in some cases pieces fell out. These plates were, therefore, condemned and cut out.

So far the cracks had been entirely confined to the flanged portion of the garboard strakes, but during the cutting out of the rivets connecting the garboard plates to B strake several plates in B strake also cracked. A new survey was then made by Lloyd's and pieces cut from the plates in the ship rolled from different charges were tested. All of the tests proved satisfactory to the surveyor, but after these tests the astonishing

fact developed that one blow of a heavy hammer was sufficient to knock large pieces out of flanged portions of the shell plates. The net result of this survey was that all plates rolled from the same charge as the garboard strakes, together with several plates which had bad surface defects, were condemned. The latter had been filled with some composition which fell away during riveting. One plate had two large holes which had been filled up. A 6-inch feeler pushed into these holes penetrated 1½ inches, while the makers' brand was not 12 inches away on the same side of the plate.

The condemned plates, twelve in number, were in various strakes of the shell plating, and during the removal, in spite of the care taken, the trouble spread over the whole of the bottom of the vessel. As an example, one of the plates in E strake, ½ inch thick, which had been punched, sheared, bent, riveted and calked without any trouble, had to have the rivets in its lower edge removed in order to take out a plate in D strake. When the rivets were center-punched preparatory to drilling them out, the plate in E strake cracked, although it was not actually touched by the hammer, which was only an ordinary hand hammer. In the edge of this plate, when all the rivets had been drilled out and driven, sixteen cracks were found.

This plate was afterwards examined by Lloyd's surveyors, when it was plainly shown that one blow with a sledge hammer was sufficient to cause cracks to appear between the edge and butt rivet holes; but, although the hammering was continued, not a single crack would travel far into the plate away from the rivet holes. The failure still continuing, a new survey was made by two of Lloyd's surveyors, one of whom had previously tested the steel. Tests were made, but these, like all the others, failed to throw the slightest light on what was really wrong with the steel. The treatment of the steel was criticised, but it was subsequently established that the workmanship in the shipyard was above reproach.

After these tests had been made the surveyors maintained that as the steel passed the tests laid down in the rules, they were not justified in blaming the material unless a test could be devised which would in their presence prove the steel to be at fault. Meanwhile, the author of the paper, who had made scores of tests with this object in view, had found one that gave the required results, and which may prove useful to others in like circumstances.

This test consisted of placing a butt strap which had not been worked into the vessel and a plate of material made in a Swedish steel works in a mixture of snow and salt, which reduced the temperature of the plates to —4 degrees F. The plates were then quickly punched on one side for double riveting and riveted together, care being taken that the edges of the plates opposite the holes were kept cool with ice in order to represent as closely as possible the conditions when working the steel in a low temperature. After the plates were allowed to cool down the rivets were driven out, during which operation the plate from the butt strap cracked badly and a piece fell out. The plate from the Swedish steel works, however, did not show the least signs of cracks.

From another plate two strips 2⅜ inches wide were cut and four holes ⅝-inch diameter and 4½ diameters apart were punched and the strips riveted together. The same was done with two similar strips, but with drilled holes, the temperature at the time being about 39.2 degrees F. After cooling, the strips were bent through an angle of 25 degrees and the outer strip was found to be broken right through in the punched sample and cracks were found in the drilled sample.

These butt-strap tests were sufficient to convince all concerned that the material was really entirely spoiled by riveting, and therefore the builders were relieved from all blame, the workmanship being characterized as "first class." A further proof that the material was bad was the very

conclusive one of the new Swedish material, which, although worked in the same way, gave no trouble at all.

From the foregoing, it is evident that the ordinary methods of testing in this particular case were insufficient to detect the poor quality of the material which was being used, and dependence upon such tests under like conditions certainly seems to place shipbuilders and others in a position in which they run grave risks. The test made by Mr. Wilson in which a mixture of snow and salt was used certainly detected the fault in this particular steel, and is therefore worthy of consideration. Subsequent tests of samples of this steel made by Professor J. O. Arnold of the University of Sheffield, showed that from a chemical point of view the steel was not of very good quality, and that it appeared to have been overheated in the manufacture and gravely injured by the operation.—"International Marine Engineering."

FRACTURE OF PLATES ON U. S. S. *O'BRIEN.*

From Report of Inspector of Machinery.

The U. S. S. *O'Brien* had nearly completed her standardization trials off the Delaware Breakwater on December 21, 1914, when it was found that a plate on the starboard side of the vessel, below the load water line, was ruptured. The after compartment of the vessel below the deck flat was filled with water, thereby increasing the displacement and seriously retarding the speed of the vessel. Subsequent examination in dry dock showed that the B and C plates on both sides, abreast the propellers, were cracked, and a piece on the starboard side torn out and bent backward.

The starboard plate cracked near the edge of the angle iron of frame 162 for a distance of 22 inches, and gradually bent outward and rearward by the pressure of water due to speed of the vessel. The crack was quite regular and parallel with the edge of the flange and clear of the rivet holes of the seam, as shown by plates A and B. Plate C shows how the cracks progress in the early stages.

These cracks were probably caused by the vibration of the vessel and the panting of the hull plates; the fractures occurring directly next to a frame stiffened by a bulkhead at that point.

EPICASSIT.

A RECENT GERMAN INVENTION FOR COATING IRON AND STEEL WITH TIN, LEAD, ZINC, ETC., BY PAINTING AND EXTERNAL HEATING.

By Charles H. Proctor.

Epicassit is a recent German invention for coating iron or steel with pure tin, tin and lead or a purely rust-free coating of tin, lead and zinc. In consists of a very finely-powdered metal which is mixed with an oil that acts as a flux. It is painted upon the metal surface in the ordinary manner with a brush and then heat is applied to the surface until the applied metal melts, when it becomes thoroughly amalgamated with the surface metal producing an extremely firm and durable coating.

Epicassit does not run when being melted, so that articles of any shape or size, in any position, can be treated without being dismantled or removed from their permanent positions.

U. S. T. B. D. "O'BRIEN"—STARBOARD SIDE.

Forward end of break is vertical on frame 162, about 22 inches high, and extends aft 15 inches on top and 9 inches on bottom. The top is between the 9-foot and 10-foot water lines and the bottom is between the 7-foot and 8-foot water lines. The break apparently started with the vertical rupture at frame 162. The forward end of break is about in line with center of propeller. The distance from the hull, at the break, to the propeller tip circle is about 18 inches.

U. S. T. B. D. "O'BRIEN"—PORT SIDE.

Vertical crack through plate on frame 162 at about same height as break on starboard side. The rag shown in the crack was drawn in as the water ran out in the dry dock.

This is one of the great advantages of Epicassit. The material will fill a distinct field and will prove of much value in the metal trades. It will enable any one to coat a metal surface with tin or its alloys in a very efficient manner, and is as economical as the older method of immersing in molten metal. Manufacturers who have occasionally to coat articles with tin or its alloys will appreciate the value of the material, as no special apparatus is required or special workman needed to produce results with the method.

Epicassit comes in several alloys, but the same oil flux is used for all.

A. Grade consists of pure Banca tin, guaranteed free from lead.

B. Grade alloy consists of two parts tin and one part lead.

C. Grade alloy consists of two parts lead and one part tin.

(C.1) Grade consists of 95 per cent. lead and 5 per cent. tin.

E. Grade consists of 35 per cent. tin, 15 per cent. lead and 50 per cent. zinc; this grade produces an entirely rust-free protective coating for articles of iron and steel.

These various grades enable the manufacturer to coat articles with any protective coating desired, from pure tin to a combination of tin-lead and zinc without any special plant. Hundreds of manufacturers in Germany are using the material. Every plumber in the United States will appreciate the value of the material. Because of the simplicity of application it will enable them to do work that heretofore they would have to refuse because of not being properly equipped to coat articles to be retinned or coated with the other alloys mentioned.

Grade A can be used where any article or part is desired to be tinned. Parts of machinery used in producing articles of food, milk cans, dairy and brewery machinery, drinking-water receptacles, heating and cooling tubes, and apparatus of all sizes in breweries, bakeries, meat packing, fruit preserving, candy making utensils and machines.

Grade B.—For any articles that come in contact with food products, for tinning bearings, as a solder for switchboard and overhead trolley connections in electric railroad operations.

Grade C.—For parts of accumulators as a protective coating against acid fumes, iron and steel window frames, textile dyeing, printing, bleaching and spinning-mill machinery, and parts can be coated to advantage as well as calendars in paper mills.

C1. Grade.—For any articles previously coated with lead gives a protective coating for metal parts coming in contact with acids equal to pure lead.

E Grade.—Gives a more perfect protective coating than any form of galvanizing against oxidation and sea water. It is particularly recommended for galvanizing articles which are to be bent, stamped, cut or corrugated. An absolute rust-protective coating for bridges, cranes, etc. When "E." Epicassit is applied before assembling, girders, plates or any other surface can be coated in or out of position. Its advantages are enumerable over any method now in vogue for the purpose of coating metals from an external source.

Epicassit is now obtainable in this country from Hess and Son, 1031 Chestnut Street, Philadelphia, Pa.—"The Metal Industry."

NAVAL VESSELS.

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1915.		Per ct. Hull com. Feb. 1, 1915.
						Jan. 1	Feb 1.	
	BATTLESHIPS:							
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	87.19	89.40	84.6
37	*Oklahoma*	New York S. Co	Reciprocating	2	20.5	88.65	90.72	87.2
38	*Pennsylvania*	Newport News Co	Cur. trb. grd. cr.	4	21	48.73	51.98	67.7
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	27.92	31.34	48.1
40	*California*	Navy Yard, N. Y.	...	4	21	...	...	...
41	*Mississippi*	Newport News Co	Cur. trb. grd. cr.	4	21	...	1.23	6.0
42	*Idaho*	New York S. Co	Pars. trb. grd. cr.	4	21	...	1.32	9.9
	DESTROYERS:							
45	*Downes*	New York S. Co	Curtis trb. & rec.	2	29	98.15	99.41	99.2
51	*O'Brien*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	95.61	95.90	93.3
52	*Nicholson*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	89.56	91.99	90.2
53	*Winslow*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	84.65	86.65	87.0
55	*Cushing*	Fore River S. Co.	Curtis trb. grd. cr.	2	29	83.75	89.69	84.0
56	*Ericsson*	New York S. Co	Pars. trb. & rec.	2	29	90.22	93.39	92.8
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	32.07	40.58	33.2
58	*Conyngham*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	38.37	41.42	54.3
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	36.10	39.20	50.8
60	*Wadsworth*	Bath Iron Works	Pars. trb. gearing.	2	30	81.77	87.14	78.6
61	*Jacob Jones*	New York S. Co	Pars. trb. grd. cr.	2	29.5	54.74	61.23	52.2
62	*Wainwright*	New York S. Co	Pars. trb. grd. cr.	2	29.5	56.51	62.29	51.7
63	...	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	...	3.03	6.4
64	...	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	...	3.03	6.4
65	...	Bath Iron Works	Pars. trb. grd. cr.	2	30	...	...	...
66	...	Bath Iron Works	Pars. trb. grd. cr.	2	30	...	...	...
67	...	Wm. Cramps & Sons	Pars. trb. grd. cr.	2	29.5	...	...	...
68	...	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	...	...	...
	FUEL SHIPS:							
13	*Kanawha*	Navy Yard, Mare Isl'd	Reciprocating	2	14	80.60	88.00	93.9
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	31.59	33.30	73.9
	SUBMARINES:							
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	84.00	85.00	86.7
40	*L-1*	Fore River S. Co.	Diesel-New Lond	2	14	93.90	95.77	89.6
41	*L-2*	Fore River S. Co.	Diesel-New Lond	2	14	93.42	95.47	84.6
42	*L-3*	Fore River S. Co.	Diesel-New Lond	2	14	92.87	94.84	75.5
43	*L-4*	Fore River S. Co.	Diesel-New Lond	2	14	91.78	94.15	76.0
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	4.50	5.03	58.2
45	*L-6*	Lake Long Beach, Cal.	Diesel-Sulzer	2	14	4.50	4.98	52.7
46	*L-7*	Lake Long Beach, Cal.	Diesel-Sulzer	2	14	4.50	4.98	51.6
47	*M-1*	Fore River S. Co.	Diesel-New Lond	2	14	86.30	88.34	63.4
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	...	...	3.0
49	*L-9*	Fore River S. Co.	Diesel-New Lond	2	14	66.66	73.13	50.0
50	*L-10*	Fore River S. Co.	Diesel-New Lond	2	14	59.34	71.76	45.6
51	*L-11*	Fore River S. Co.	Diesel-New Lond	2	14	33.96	48.53	33.2
	SUBMARINE TENDER:							
2	*Bushnell*	Seattle Con. & D. D. Co.	Pars. trb. gearing	1	14	60.73	66.69	83.3
	DESTROYER TENDER:							
2	*Melville*	New York S. Co.	Pars. trb. gearing	1	15	69.39	72.00	78.3
	Transport	Navy Yard, Phila.	Reciprocating	2	14	...	1.85	3.4
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	1.03	1.33	15.8
	PANAMA COLLIERS:							
	Ulysses	Md. Steel Co.	Reciprocating	2	14	80.13	87.96	87.2
	Achilles	Md. Steel Co.	Reciprocating	2	14	64.35	75.19	77.1
	Oil Barges 8 and *9*.	Navy Yard, Mare Isl'd	...	...	...	41.70	69.30	...
	Oil Barge 7	Navy Yard, Norfolk	...	...	...	82.00	85.00	...

U. S. S. *DOWNES.*

The U. S. S. *Downes* successfully completed her preliminary trials off the Delaware Breakwater, January 5, 1915. An average speed of 29.071 knots per hour was maintained on the four-hour full-speed trial. Delivery of vessel has been authorized, Navy Yard, Philadelphia, Pa.

U. S. S. *O'BRIEN.*

The U. S. S. *O'Brien* successfully completed her preliminary trials (second trial) off the Delaware Breakwater, February 6, 1915. An average speed of 29.15 knots per hour was maintained in the four-hour full-speed trial.

SUBMARINE TENDER *FULTON.*

The standardization trials were made on the Provincetown course, and the highest speed on the average of five runs was 12.78 knots while the contract called for but 12¼ knots. The anchor tests and maneuvering trials were equally successful, particularly in reversing at full speed and bringing the vessel to a dead stop. The fuel test at cruising speed proved the *Fulton's* motor plant to be economical, while the working of the engine and all auxiliaries was perfect. Her fuel-carrying capacity gives the *Fulton* a cruising radius of 10,000 miles and, as compared with the fuel consumption of a steam vessel, the *Fulton,* on a given amount of fuel, could make four to five times the distance that a coal-fired vessel could. To this advantage must also be added the economy in fuel bills and the saving in the number of men necessary to handle the machinery.

The "Rudder" for January contained a detailed description of this vessel together with outboard and inboard profile and accommodation, diagrams and other illustrations in connection therewith.—Exchange.

NAVAL REINFORCEMENTS.

The additions which have been, and will be made to the belligerent fleets since the commencement of hostilities demonstrate the fact that the longer the war lasts the greater will be the superiority of the British over the German Navy. The following is a list of all classes of ships so far as is known which have been or will be completed before the end of 1915.

BRITAIN.

BATTLESHIPS COMPLETED.—(1) *Agincourt,* 27,500 tons, fourteen 12-in. guns; (2) *Erin,* 23,000 tons, ten 13.5-in. guns; (3) *Benbow,* 25,000 tons, ten 13.5-in. guns; (4) *Emperor of India,* 25,000 tons, ten 13.5-in. guns. BATTLESHIPS BUILDING.—(5) *Queen Elizabeth,* 27,500 tons, eight 15-in. guns; (6) *Warspite,* 27,500 tons, eight 15-in. guns; (7) *Valiant,* 27,500

tons, eight 15-in. guns; (8) *Barham,* 27,500 tons, eight 15-in. guns; (9) *Malaya,* 27,500 tons, eight 15-in. guns; (10) *Royal Sovereign,* 26,000 tons, eight 15-in. guns; (11) *Royal Oak,* 26,000 tons, eight 15-in. guns; (12) *Revenge,* 26,000 tons, eight 15-in. guns; (13) *Ramillies,* 26,000 tons, eight 15-in. guns; (14) *Canada,* formerly *Almirante Latorre* (building for Chili), 28,000 tons, ten 14-in. guns. BATTLE CRUISERS COMPLETED.—*Tiger,* 28,000 tons, eight 13.5-in. guns. CRUISERS (nearly all completed).—(1) *Arethusa,* 3,520 tons, two 6-in. and six 4-in. guns; (2) *Undaunted,* 3,520 tons, two 6-in. and six 4-in. guns; (3) *Aurora,* 3,520 tons, two 6-in. and six 4-in. guns; (4) *Galatea,* 3,250 tons, two 6-in. and six 4-in. guns; (5) *Inconstant,* 3,520 tons, two 6-in. and six 4-in. guns; (6) *Royalist,* 3,520 tons, two 6-in. and six 4-in. guns; (7) *Penelope,* 3,520 tons, two 6-in. and six 4-in. guns; (8) *Phaeton,* 3,520 tons, two 6-in. and six 4-in. guns; (9) *Calliope,* 3,800 tons, three 6-in. and six 4-in. guns; (10) *Caroline,* 3,800 tons, three 6-in. and six 4-in. guns; (11) *Carysfort,* 3,800 tons, three 6-in. and six 4-in. guns; (12) *Champion,* 3,800 tons, three 6-in. and six 4-in. guns; (13) *Cleopatra,* 3,800 tons, three 6-in. and six 4-in. guns; (14) *Comus,* 3,800 tons, three 6-in. and six 4-in. guns; (15) *Conquest,* 3,800 tons, three 6-in. and six 4-in. guns; (16) *Cordelia,* 3,800 tons, three 6-in. and six 4-in. guns. MONITORS (completed).—(1) *Humber,* 1,200 tons, two 6-in. and two 4.7-in. guns; (2) *Severn,* 1,200 tons, two 6-in. and two 4.7-in. guns; (3) *Mersey,* 1,200 tons, two 6-in. and two 4.7-in. guns; (4) Unnamed, 4,900 tons, two 9.4-in., four 6-in. and six 4-in. guns; (5) Unnamed, 4,900 tons, two 9.4-in., four 6-in. and six 4-in. guns. DESTROYERS (Flotilla Leaders) completed.—*Broke, Faulkner,* 1,600 tons, six 4-in. guns; *Marksman, Lightfoot,* and probably more purchased. DESTROYERS completed.—*Lance, Laverock, Leonidas, Milne, Moorsom, Mastiff, Lookout, Lucifer, Meteor, Minos, Miranda, Manly, Morris, Mansfield, Myngs, Murray, Matchless.* SUBMARINES.—Unknown.

GERMANY.

BATTLESHIPS.—(1) *König,* 25,800 tons, ten 12-in. guns; (2) *Kronprinz,** 25,800 tons, ten 12-in. guns; (3) *Markgraf,* 25,800 tons, ten 12-in. guns; (4) *Grosser Kurfürst,* 25,800 tons, ten 12-in. guns. BATTLE CRUISERS.—(1) *Derflinger,* 26,600 tons, eight 12-in. guns; (2) *Lutzow,* 26,600 tons, eight 12-in. guns; (3) *Erz. Hertha,** 26,600 tons, eight 12-in. guns; (4) ex. Greek *Salamis,** 19,500 tons, eight 14-in. guns. CRUISERS.—(1) *Karlsruhe,* (2) *Rostock,* (3) *Grandenz,* (4) *Enz. Niobe,** (5) *Enz. Gefion,** (6) *Enz. Gazele,** (7) Unnamed, 4,500 tons; (8) Unnamed, 4,500 tons, eight 5-in. guns (building for Russia). DESTROYERS AND SUBMARINES.—Uncertain.

FRANCE.

BATTLESHIPS.—(1) *France,* 23,467 tons, twelve 12-in. guns; (2) *Paris,* 23,467 tons, twelve 12-in. guns; (3) *Bretagne,** 23,550 tons, ten 13.4-in. guns; (4) *Lorraine,** 23,550 tons, ten 13.4-in. guns; (5) *Provence,** 23,550 tons, ten 13.4-in. guns. DESTROYERS.—About nine completing.

JAPAN.

BATTLESHIPS.—*Fu So,** 30,600 tons, twelve 14-in. guns. BATTLE CRUISERS.—*Hi-Yei,* 27,500 tons, eight 14-in. guns; *Haruna,** 27,500 tons, eight 14-in. guns; *Kirishima,** 27,500 tons, eight 14-in. guns. DESTROYERS.—Two.

---

* Building.

RUSSIA.

BATTLESHIPS.—(1) *Pollava,* 23,300 tons, twelve 12-in. guns; (2) *Petropavlovsk,* 23,300 tons, twelve 12-in. guns; (3) *Sevastopol,* 23,300 tons, twelve 12-in. guns; (4) *Ganfoot,* 23,500 tons, twelve 12-in. guns; (5) *Ekaterina II.,** 22,500 tons, twelve 12-in. guns; (6) *Imperatrza Maria,** 22,500 tons, twelve 12-in. guns; (7) *Alexander III.,** 22,500 tons, twelve 12-in. guns. CRUISERS.—(1) *Svietlana,* 6,750 tons, twelve 6-in. guns; (2) *Greig,* 6,750 tons, twelve 6-in. guns; (3) *Bootakof,* 6,750 tons; twelve 6-in. guns; (4) *Spiridoff,* 6,750 tons, twelve 6-in. guns (all not likely to be finished during the war).

AUSTRIA.

BATTLESHIPS.—*Szent Istvan,** 20,000 tons, twelve 12-in. guns.

NEW JAPANESE DESTROYERS.

The two ocean-going torpedo-boat destroyers, *Kawakaze* and *Urakase,* which Yarrow and Co., Ltd., will shortly launch at Glasgow for the Japanese Government, are of exceptional engineering interest, because while in each the main propelling machinery is to consist of an installation of steam turbines, each will also have for cruising purposes, two Diesel engines of 1,200 B.H.P. The twin turbines will each be of 12,000 H.P., and when the vessels are running at cruising speeds the twin screws will be driven by the oil engines, which it is understood will operate the propellers by means of solid shafts working inside tubular shafts.

The Diesel engines, which are being supplied by the Burmeister and Wain (Diesel system) Oil Engine Co., of Glasgow, are of the six-cylinder four-cycle type, and they are designed to give the vessels a speed of from 13 to 14 knots. When the turbines are in use the speed will be about 34 knots. The advantage of the cruising machinery lies in the great saving in fuel which will be possible and the rapidity with which the vessels can be got under way.—"Page's Engineering Weekly."

STRENGTH OF NAVAL POWERS.

The data given below as to the relative strength of the principal naval powers were furnished by the Office of Naval Intelligence, Navy Department, as of date July 1, 1914. Owing to the state of war in Europe the Office of Naval Intelligence was unable to furnish any definite information of the relative strength of the principal naval powers of the world later than July 1, 1914.

The figures given are incorporated in the Navy Year Book, just issued as a Senate document, which is a compilation of annual Naval Appropriation laws from 1883 to 1914, prepared by J. D. Knight, Secretary of the Senate Committee on Naval Affairs. It is shown that the United States Navy, on the basis of ships completed on July 1, 1914, was third among the navies of the world in respect to tonnage. Great Britain was first and Germany second. When vessels under construction on July 1 are completed the United States Navy will be in fourth place, surpassed by Great Britain, Germany and France. Figures are lacking, however, to show the progress in naval increase among belligerents in the present war in Europe since its commencement, and it is well known that both England and Germany have been increasing their naval power as rapidly as possible.

* Building.

WARSHIP TONNAGE OF THE PRINCIPAL NAVAL POWERS.

[Number and displacement of warships built and building, of 1,500 or more tons, and torpedo craft of more than 50 tons.]

Type of vessel.	Great Britain (including colonial vessels).				Germany.				United States.				France.			
	Built.		Building.		Built.		Building.		Built.		Building.		Built.		Building.	
	No.	Tons.	No.	Tons (estimated).	No.	Tons.	No.	Tons (estimated).	No.	Tons.	No.	Tons (estimated).	No.	Tons.	No.	Tons (estimated).
Battleships (dreadnaught type) [1]	20	423,350	16	421,750	13	285,770	7	187,164	8	189,650	4	117,800	4	92,368	8	193,656
Battleships (predreadnaught) [2]	40	589,385	...	...	20	242,800	...	...	22	309,282	...	...	18	262,675	...	...
Coast-defense vessels [3]	...	...	...	...	2	8,168	...	...	4	12,900	...	...	1	8,800	...	...
Battle cruisers [4]	9	187,800	1	28,500	4	88,749	4	112,000	...	...	...	...	...	...	...	...
Armored cruisers	34	406,800	...	...	9	94,245	...	...	11	149,295	...	...	20	201,724	...	...
Cruisers [5]	74	382,815	17	67,000	41	150,747	5	26,900	14	66,410	...	...	9	46,095	...	...
Torpedo-boat destroyers	167	125,850	21	21,770	130	67,094	24	14,400	51	35,068	11	11,956	84	35,812	3	2,653
Torpedo boats	49	11,488	...	...	...	...	...	...	13	2,528	...	...	135	13,426	...	...
Submarines	75	30,362	22	17,236	27	14,140	18	14,400	30	...	[7]19	...	64	27,940	22	14,766
Total tons built and total tons building	...	2,157,850	...	556,256	...	951,713	...	354,864	...	765,133	...	129,756	...	688,840	...	211,075
Total tons built and building	2,714,106				1,306,577				894,889				899,915			

Type of vessel.	Japan.				Russia.				Italy.				Austria-Hungary.			
	Built.		Building.		Built.		Building.		Built.		Building.		Built.		Building.	
	No.	Tons.	No.	Tons (estimated).	No.	Tons.	No.	Tons (estimated).	No.	Tons.	No.	Tons (estimated).	No.	Tons.	No.	Tons (estimated).
Battleships (dreadnaught type) [1]	2	41,600	4	122,400	...	...	7	159,409	3	62,644	7	187,150	3	60,030	4	93,510
Battleships (predreadnaught) [2]	13	191,380	...	...	7	98,750	...	...	8	96,100	...	...	6	74,613	...	...
Coast-defense vessels [3]	2	9,086	...	...	2	10,380	...	...	...	...	...	...	6	41,700	...	...
Battle cruisers	2	55,000	2	55,000	...	...	4	128,000	...	...	...	...	...	...	...	...
Armored cruisers	13	138,483	...	...	6	63,500	...	...	9	74,020	...	...	2	13,380	...	...
Cruisers [5]	13	57,915	...	...	9	52,845	8	53,600	6	18,830	2	4,888	5	13,815	5	21,216
Torpedo-boat destroyers	50	20,487	2	1,676	91	36,748	44	53,664	36	16,807	15	14,203	18	9,450	...	...
Torpedo boats	27	3,017	...	...	14	2,132	...	...	68	11,584	2	272	39	6,852	24	5,886
Submarines	13	2,672	2	1,200	30	6,506	19	13,284	19	5,475	8	5,842	6	1,686	6	5,370
Total tons built and total tons building	...	519,640	...	180,276	...	270,861	...	407,957	...	285,460	...	212,355	...	221,526	...	125,982
Total tons built and building	699,916				678,818				497,815				347,508			

Warship Tonnage Table Notes.

[1] Battleships having a main battery of all big guns. (11 inches or more in caliber.)
[2] Battleships of (about) 10,000 or more tons displacement, whose main batteries are of more than one caliber.
[3] Includes smaller battleships and monitors.
[4] Armored cruisers having guns of largest caliber in main battery and capable of taking their place in line of battle with the battleships. They have an increase of speed at the expense of carrying fewer guns in main battery and a decrease in armor protection.
[5] All unarmored warships of more than 1,500 tons are classed as cruisers. Scouts are considered as cruisers in which battery and protection have been sacrificed to secure extreme speed. The word "protected" has been omitted, because all cruisers except the smallest and oldest now have protective decks.
[6] Does not include *Idaho* and *Mississippi*, recently sold, or ships of current program for which contracts have not been awarded
[7] Includes three submarines authorized in 1913; contract for fourth not yet awarded.

ACTIVE PERSONNEL.

Rank.	England.	Germany.	United States.	France.	Japan.	Russia.	Italy.	Austria-Hungary.
Admirals of the fleet	3	2	[1] 1	...	2	...	...	...
Admirals	12	6	...	...	6	12	1	1
Vice admirals	22	12	...	15	19	20	10	2
Rear admirals	58	22	[2] 25	30	38	21	19	15
Captains and commanders	702	154	212	360	270	346	137	80
Other line officers	2,508	2,220	1,680	1,419	1,965	1,378	753	558
Midshipmen at sea	639	448	...	77	119	...	73	175
Engineer officers	837	577	...	505	811	538	326	164
Medical officers	593	340	336	[3] 390	[4] 364	286	[3] 259	84
Pay officers	750	276	231	211	388	...	228	[5] 224
Naval constructors	122	162	75	187	[6] 135	519	107	[7] 141
Chaplains	147	30	24	...	...	85	...	11
Warrant officers	2,740	3,183	867	[8] 147	1,569	...	1,340	387
Enlisted men	119,597	65,797	52,566	60,505	50,050	49,258	36,660	[9] 17,689
Marine officers	465	[10] 177	341	...	...	...	...	...
Enlisted men (marines)	* 21,414	[10] 5,791	9,915	...	...	...	...	...
Total	150,609	79,197	66,273	63,846	55,736	52,463	39,913	19,531

Personnel Table Notes.

[1] The Admiral of the Navy.
[2] The United States now has, temporarily, as extra numbers, due to promotion for war service and to officers restricted by law to engineering duty only on shore only, 6 flag officers, 20 captains, 9 commanders, 6 lieutenant commanders and 1 lieutenant.
[3] Includes pharmacists.
[4] Includes pharmaceutical officers.
[5] Includes 21 officers of the Judge Advocate's Corps.
[6] Includes 50 ordnance and 10 hydrographic engineers.
[7] Includes 4 hydrographic engineers.
[8] Includes adjutants principaux; does not include premier maitres and maitres.
[9] Includes 4,000 recruits for 42 days.
[10] Marine infantry and seaman artillery.
* Includes 3,130 men of the Coast Guard.
NOTE.—In the table published Dec. 1, 1913, the number of captains and commanders given was 356 and other line officers 1,881. This apparent discrepancy was due to the inclusion of 213 Korvetten Kapitane (lieutenant commanders) with the captains and commanders. In the above table the Korvetten Kapitane (226) are included with the other line officers. Under Italy the number of vice admirals given was 18; this was a typographical error and should have been 8.

The following vessels are not included in the tables above:

Ships over twenty years old from date of launch, unless they have been reconstructed and re-armed within five years; torpedo craft over fifteen years old; those not actually begun or ordered, although authorized; transports, colliers, repair ships, torpedo depot ships, or other auxiliaries; vessels of less than 1,500 tons, except torpedo craft of less than fifty tons.

Vessels undergoing trials are considered as completed.

England has no continuing shipbuilding policy, but usually lays down each year four or five armored ships with a proportional number of smaller vessels.

Germany has a continuing shipbuilding program, governed by a fleet law authorized by the Reichstag. For 1913 there are authorized one battleship,

one battle cruiser, two cruisers, twelve destroyers. Eventual strength to consist of forty-one battleships, twenty armored cruisers, forty cruisers, 144 destroyers, seventy-two submarines.

Japan authorized $78,837,569 to be expended from 1911 to 1917 for the construction of war vessels.

Russian shipbuilding program provides for the completion by 1918 of four battle cruisers, eight small cruisers, thirty-six destroyers and eighteen submarines.—"Army and Navy Journal."

OBITUARY.

The death of Charles Ward on January 17, 1915, removed an engineer who had played a prominent part in marine engineering and whose efforts were always given to improvement and increase of efficiency. Mr. Ward was born in Seamington, England, March 5, 1841, and came to America in 1871. He had been trained as a gas engineer in England, and almost his first work in America was in Charleston, West Virginia, where he installed the first gas works and later became the superintendent and general Manager of the company. From this time Charleston became his home.

Outside of local prominence in western-river steamboating, Mr. Ward came into prominence before the engineers of the country when the late Admiral Melville invited the makers of water-tube boilers to compete for supplying the boilers for the coast defense vessel *Monterey*. Mr. Ward's boiler was tested in 1890, and four boilers of this type were installed in the *Monterey*, the first installation of water-tube boilers in a large war vessel.

Before this time smaller water-tube Ward boilers of different design had been used in steam launches of the United States Navy, and this type is being used at the present time.

Mr. Ward was also a pioneer in the use of screw propellers on western-river steamboats in the effort to reduce waste and increase efficiency as compared with the time-honored sternwheel boats.

Mr. Ward was a man of agreeable personality who inspired confidence and respect in all who were associated with him. His business, which was at first carried out in his own name, later became The Charles Ward Engineering Works, which, since his serious illness some eight or ten years ago, has been carried on by his son, Mr. Charles E. Ward.

ASSOCIATION NOTES.

At a meeting of the Society held at the Navy Department December 30, 1914, the ballots cast for candidates for officers of the Society for the year 1915 were counted and the following declared elected:

President, Captain S. S. Robison, U. S. Navy.

Secretary-Treasurer, Lieutenant A. T. Church, U. S. Navy.

Members of Council: Captain B. C. Bryan, U. S. Navy; Engineer-in-Chief C. A. McAllister, U. S. C. G.; Commander U. T. Holmes, U. S. Navy.

Commander U. T. Holmes having been detached from duty in Washington, tendered his resignation as a member of the Council. Lieutenant Commander J. O. Richardson, U. S. N., was elected to fill the vacancy.

Lieutenant Commander H. C. Dinger, U. S. Navy, retiring Secretary-Treasurer, has rendered his financial statement in accordance with the By-Laws of the Society. The books were audited and found to be correct. On December 31, 1914, the total assets of the Society amounted to $10,167.50. There were no liabilities.

The numerical strength of the Society was as follows:

Members,	544
Associates,	331
Subscribers,	366
Exchanges,	73
Total,	1,314

The annual banquet of the Society was held at the Army and Navy Club in the City of Washington, D. C., on the evening of February 20, 1915. About 150 members and

guests were present, among whom were several members of both branches of Congress. The occasion was a most enjoyable one. Mr. H. L. Ferguson, of the Newport News Shipbuilding Company, acted as toastmaster and the following gentlemen responded to toasts:

Rear Admiral Bradley A. Fiske, U. S. Navy.

Senator CHARLES F. JOHNSON, of Maine.

Mr. WALTER M. MCFARLAND, of the Babcock & Wilcox Company.

Lieutenant Commander L. C. RICHARDSON, U. S. Navy.

The committee in charge was as follows:

Engineer-in-Chief C. A. MCALLISTER, U. S. C. G.

Lieutenant Commander H. C. DINGER, U. S. Navy.

Lieutenant Commander J. O. RICHARDSON, U. S. Navy.

Lieutenant A. T. CHURCH, U. S. Navy.

THE FOLLOWING MEMBERS AND ASSOCIATES have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Babbitt, Herbert S., Lieutenant, U. S. Navy.
Baer, Joseph, Lieutenant, U. S. Navy.
Bassett, Prentiss P., Lieutenant, U. S. Navy.
Bradley, Willis W., Jr., Lieutenant, U. S. Navy.
Gayler, Ernest R., Civil Engineer, U. S. Navy.
Dixon, Virgil J., Lieutenant, U. S. Navy.
Hart, Thomas C., Lieutenant Commander, U. S. Navy.
Harvey, Urban, Lieutenant of Engineers, U. S. C. G.
Hill, Ellis Reed, 3d Lieutenant of Engineers, U. S. C. G.
Jennings, John C., Lieutenant, U. S. Navy.
Kingman, Howard F., Lieutenant, U. S. Navy.
Maxfield, Louis H., Lieutenant, U. S. Navy.
Owens, Charles T., Lieutenant Commander, U. S. Navy.
Peyton, Paul J., Lieutenant, U. S. Navy.
Prall, Whitney M., 2d Lieutenant of Engineers, U. S. C. G.
Smith, Harold T., Lieutenant, U. S. Navy.
Williams, Raleigh C., Lieutenant, U. S. Navy.

ASSOCIATES.

Broström, P. Daniel, Göteborg, Sweden.

Cornbrooks, Thomas M., Chief Engineer, Maryland Steel Company, Sparrows Point, Md.

Foord, James L., Chief Inspector, The Hartford Steam Boiler Inspection and Insurance Company, 800 Royal Insurance Building, Chicago, Ill.

Keller, O. B., care of Keuffel and Esser Co., 127 Fulton St., New York City.

Oatley, Henry B., Chief Engineer Locomotive Superheater Company, 30 Church St., New York City.

Rotter, Max, Chief Engineer, Busch-Sulzer Brothers Diesel Engine Co., St. Louis, Missouri.

Shipman, Frank J., The Texas Company, 17 Battery Place, New York City.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXVII. MAY, 1915. No. 2.

HEAT TRANSMISSION AND TUBE LENGTH IN MARINE FEED-WATER HEATERS.

BY LEO LOEB, M. E., ASSOCIATE.*

In the attempt to promote the economy and efficiency of all power machinery and to add to the scientific knowledge underlying the correct design of prime movers and auxiliaries, both stationary and marine, the general subject of heat transmission between two fluids separated by a metal wall has been critically investigated in recent years. The results of such investigations, both in the United States and abroad, have enabled designing engineers to grasp the principles underlying the distribution of heating surface in boilers to such an extent that stationary boilers operating on coal fuel and mechanically stoked show overall efficiencies as high as 76.2 per cent. when generating 7.4 pounds of equivalent steam per hour per square foot of heating surface. Naval practice has progressed to the point where an oil-fired water-tube boiler develops efficiencies of 79.0 and 76.5 per cent. when generating respectively 10.3

*Professor of Marine Engineering, Post Graduate Department, U. S. Naval Academy.

and 18.7 pounds of equivalent steam per hour, discharging stack gases with little dilution and low carbon monoxide at stack temperatures of 561 and 759 degrees Fahrenheit.*

In the field of condenser practice the recent important contribution to scientific literature is " The Transmission of Heat in Surface Condensation," George A. Orrok, "American Society of Mechanical Engineers," 1910, p. 1139–1214.

Articles dealing with the economies effected by feed heating or the possibilities of increasing the economy of plants by the introduction of feed heaters are frequently seen. It is the intention in this instance, however, to develop the theory of heat transmission as applied to closed heaters operating on exhaust steam and to apply this theory with reliable data from tests of marine feed heaters to the preparation of design curves incorporating all the operating variables for a given type of heater. The results from practice naturally show less departure from the theoretical deductions than with other heat transfer apparatus, inasmuch as there is an absence of soot or scale deposits which require frequent interruptions in the case of boiler and evaporator researches, and freedom of the difficulties arising from presence of air in large volume at the pressures dealt with in experiments on condensers. Given the necessary mechanical equipment, tests may without great difficulty be conducted to determine effect of velocity, exhaust steam pressure or initial temperature as desired, since the problem resolves itself to a direct transfer of heat between condensing steam and water without, as a rule, the disturbing elements of scale or air.

I.—General Theory of Heat Transfer from Steam to Water.

When a metal tube carrying a moving column of water is exposed to the heating action of surrounding steam there occurs a physical state at each section which may be illustrated

*Oil Burning, J. J. Hyland, Lieutenant Commander, U. S. N., " Journal of American Society of Naval Engineers," May, 1914, p. 400.

by figure 1. The steam space has a constant temperature represented by the line AB. Immediately surrounding the tube is a film composed of particles of steam and water of condensation, and the resistance to heat flow therein results in a tem-

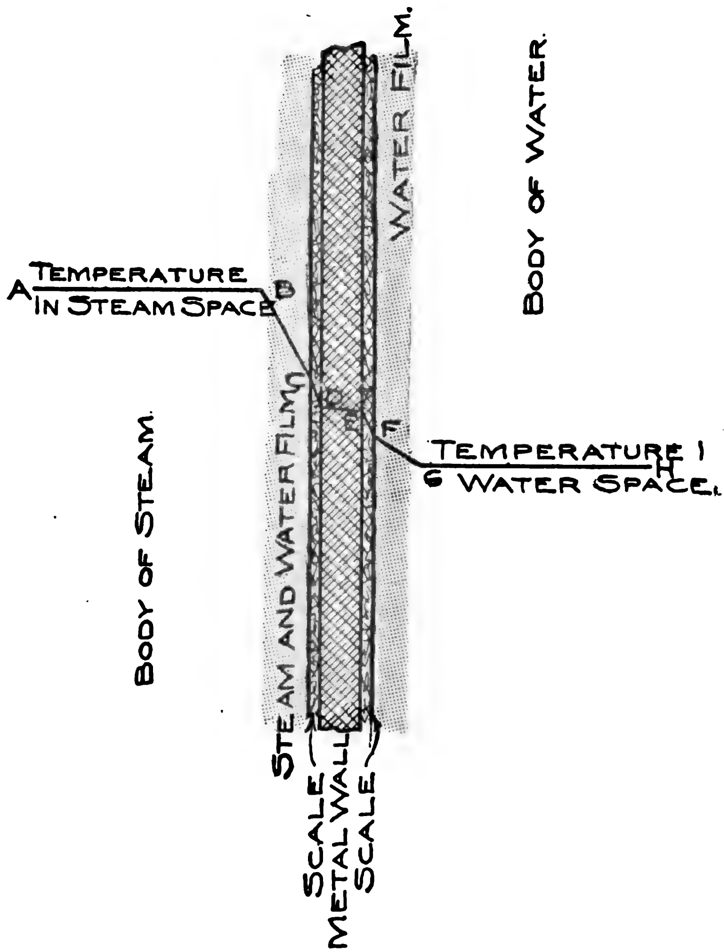


FIG. 1.

perature drop represented by the line BC; the tube wall may have a scale resulting from the process of manufacture or from the deposition of chemical salts which results in a temperature drop CD; the metal wall of the tube proper accounts for a slight temperature drop DE, while on the water side

there may again exist a thin scale and there will always be a water film with temperature drops EF and FG. The total drop in temperature is from AB in the steam space to GH in the body of the water. If the steam temperature be substantially constant throughout the heater, then the relation between steam temperature and water temperature for a single tube is represented by figure 2. The steam temperature, t_s is uniform and the water entering on the left at an inlet temperature t_i has heat added by conduction according to the temperature gradient, G, till it reaches the outlet temperature t_o. Other things being equal, the form of this temperature gradient will depend, first, upon the temperature difference between the steam and water, since temperature difference is the impelling force in heat transmission, and, secondly, upon the resistance offered by the films and metal walls.

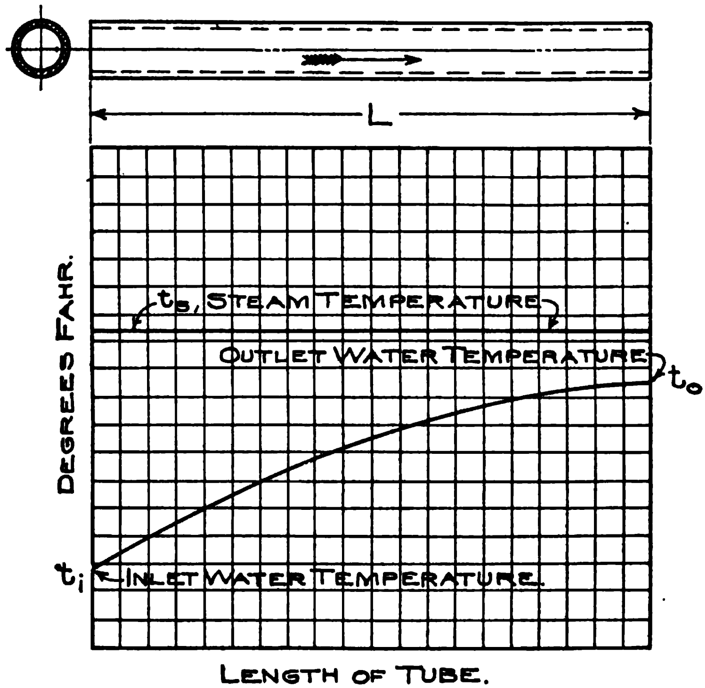


FIG. 2.

Heat resistance is best measured in terms of its reciprocal, K, thermal conductivity, the number of heat units transmitted

per unit area across a given space per unit of time. Denoting by

Q = heat transferred per hour in B.t.u.;
t_m = mean temperature difference;
S = area of heating surface in square feet;
K = rate of heat transmission in B.t.u. per hour per square foot per degree temperature difference;

then

$$Q = SKt_m.$$

II.—Thermal Resistances.

Considering first the nature of K: The total thermal resistance is dependent upon the two film resistances, scale and metal-wall resistances. Since proper preparation of the tube material and proper upkeep and operation will eliminate the scale conditions, the resistance may be reduced to three:

(a) Steam and water film;
(b) Metal wall;
(c) Water film.

The relative values of thermal conductivities as derived from data in the Smithsonian Physical Tables, 6th edition, 1914, are given below:

Material.	Conductivity in B.t.u. per hour per sq. ft. of surface per inch of thickness.
Aluminum	966.
Brass	736.6
Copper	2633.2
Wrought Iron	411.8
Steel, soft	321.9
Silver	3180.
Tin	412.7
Asbestos paper	1.247
Water	4.553
Salt solution	7.743
Air	.16476

Although this table does not include steam, the value for vapor will probably be above that of air but considerably below that of water. The values of thermal conductivity for

metals should be carefully noted. It may be observed that the transfer of heat through copper is 8.17 times as great as through steel of the same thickness, but since tube material is seldom over 1/10 inch thick, there will result 26,332 and 3,219 B.t.u. per hour per square foot per degree temperature difference for an average copper and an average steel tube. Since the usual coefficients of heat transmission will vary from 400 to 1,000 B.t.u. per hour per square foot of heating surface per degree average temperature difference between steam and water, it must be plain that the temperature difference on the two sides of the metal tube is very little; that tube material or tube thickness only slightly impedes heat flow, and that a water film .00173 inch thick will give the same resistance as a 1 inch thickness of copper tube. It is evident, therefore, that a metal tube will transmit all the heat that is presented to its surface and that the controlling resistances lie in the two films which cling to the metal surface. The condensing steam presents a wet surface, so that resistances on the two sides are much alike.

The formation of such a film is a friction effect. The microscopic irregularities of surface of the metal walls tear off particles of the fluids passing and prevent these particles from being swept along with the major current. The more completely these particles of water are dislodged and swept from the surface, the better able are other colder particles to replace them and absorb their share of heat from the surface. Hence the problem in producing high transmissions in heaters is the destruction of this water film by a scrubbing action produced by a high velocity along the heating surface. When the limit of heat transmission has been reached on the water side by a velocity that is entirely practical, the controlling resistance has passed to the steam side, and the only way to further increase heat transmission is to sweep away the film on the steam side.

III.—Temperature Differences.

A factor in the heat transfer equation of equal importance with the coefficient of conductivity is temperature difference.

This temperature difference can be increased in but one way, by raising the temperature of the heating medium, since the feed inlet is fixed in practice by the temperature of the condenser wet-pump discharge and hotwell drains. An increase in exhaust temperature is obtained by throttling the auxiliary exhaust, a method which off-hand would not seem to be conducive of best overall results. A further analysis of the situation may show this to be of some value. Direct-acting feed pumps as at present designed do not realize an economy from low back pressures, as apparently the restricted ports build up a certain minimum back pressure below which there is no reduction in steam consumption by a lower exhaust pressure. A careful analysis of this condition requires knowledge of steam consumption of pumps and blowers at varying speed and back pressures. Much test data on steam consumption of direct-acting pumps and turbine-driven blowers are on file at the Bureau of Steam Engineering, and curves of total heat consumption and boiler-heat economy could be drawn as a function of exhaust pressure. Their intersection would determine the best exhaust pressure for a given speed.

The type of temperature gradient resulting from the assumption that heat transfer is at any instant proportional to temperature difference is shown in the curve on figure 3.

In this figure let:

t = fluid temperature at any point;
l = length of tube measured from inlet;
W = weight of fluid per hour;
c = specific heat of fluid;
S = heating surface per unit of length of tube;
K = coefficient of heat transfer;
t_s = steam temperature;
t_i = inlet water temperature;
t_o = outlet water temperature;
t_m = mean temperature difference.

In a length of tube dl the fluid temperature will rise dt due to the temperature difference, $t_s - t$. The heat absorbed by the fluid in B.t.u. per hour will be

$$cWdt,$$

and the transfer across the metal wall due to temperature difference is

$$K\,(t_s - t)\,Sdl.$$

Equating these two relations

$$cWdt = K(t_s - t)\,Sdl,$$

$$\frac{dt}{t_s - t} = \frac{KSdl}{cW}.$$

Combining the constants K, S, c, W into another constant K′ and integrating:

$$\log_e (t_s - t) = \frac{KS}{cW}\,l + \text{constant},$$

which is a relation between heating surface and temperature variation in a heater tube if heat transmission is proportional to temperature difference. Between limits of inlet and outlet temperature t_i and t_o and for the entire length l', this becomes

$$\log_e \left(\frac{t_s - t_i}{t_s - t_o}\right) = K'l' = \frac{KSl'}{cW},$$

$$Q = KSl't_m,$$

and

$$Q = cW(t_o - t_i).$$

Hence

$$K\,Sl't_m = cW(t_o - t_i);$$

But

$$\frac{KSl'}{cW} = \log_e \left(\frac{t_s - t_i}{t_s - t_o}\right).$$

Hence

$$t_m = \frac{t_o - t_i}{\log_e \left(\frac{t_s - t_i}{t_s - t_o}\right)}.$$

Using the above relation between temperatures and length of tube it is possible to construct the temperature gradient when the inlet and outlet feed temperatures are known.

For the full length of tube, l', and the temperature limits t_i and t_o;

$$K'l' = \log_e\left(\frac{t_s - t_i}{t_s - t_o}\right).$$

For a fraction length, say $l = \frac{l'}{2}$.

$$K'\frac{l'}{2} = \log_e\left(\frac{t_s - t_i}{t_s - t_3}\right),$$

where t_3 is the temperature at half tube length.

$$2\log_e\left(\frac{t_s - t_i}{t_s - t_3}\right) = \log_e\left(\frac{t_s - t_i}{t_s - t_o}\right),$$

$$\log_e\left(\frac{t_s - t_i}{t_s - t_3}\right)^2 = \log_e\left(\frac{t_s - t_i}{t_s - t_o}\right).$$

Dropping the logs,

$$\left(\frac{t_s - t_i}{t_s - t_3}\right)^2 = \left(\frac{t_s - t_i}{t_s - t_o}\right),$$

$$t_s - t_3 = \sqrt{(t_s - t_i)(t_s - t_o)},$$

$$t_3 = t_s - \sqrt{(t_s - t_i)(t_s - t_o)}.$$

Similarly for other fractional lengths as $l = \frac{l'}{4}$, $t = t_4$,

$$(t_s - t_4) = \sqrt[4]{(t_s - t_i)^3(t_s - t_o)}.$$

In this way the temperature gradient for a given tube may be plotted, as the curve in figure 3, when inlet and outlet temperatures for a given steam pressure and water velocity are known, and this curve will determine the characteristics of shorter or longer heaters without knowing the numerical value of the coefficient of heat transfer.

The above analysis would be very satisfactory were it not for the fact that there exists considerable experimental evidence that heat transfer in feed heaters is not directly proportional to temperature difference but to some fractional power of the temperature difference. Tests on actual heaters where

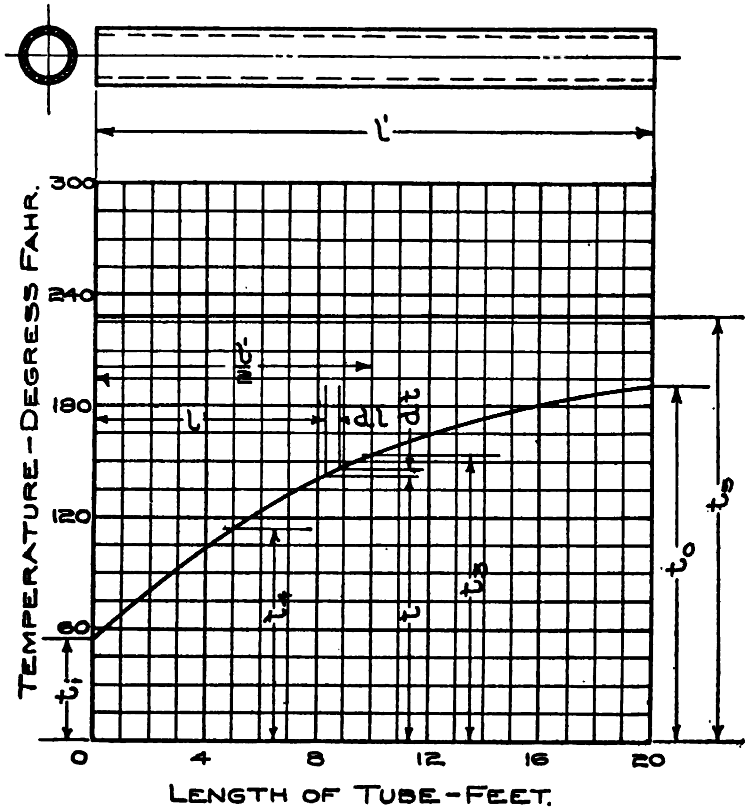


FIG. 3.

this fact appears will be discussed at a later time, but for the purpose of establishing the relation between temperature difference and heat transfer there are cited results obtained from a single-tube experimental heater tested under the direction of the writer at the Engineering Experiment Station. This heater consisted of a single ¾-inch copper tube. No. 18 B.W.G., four feet long, secured within a 4-inch pipe, which formed the steam space. The inlet could be regulated to any desired temperature by an auxiliary heater, and the temperature gradient at any given water rate was determined by noting inlet and outlet temperature at three-minute intervals for a period of 18 minutes. The inlet temperature was then raised

in the auxiliary heater to correspond closely with the average outlet temperature of the initial run and data taken as before.

Thus there could be determined the temperatures at four-foot intervals of a heater whose length could be varied to suit the temporary conditions of velocity and steam temperature. The data in Table I is from a typical set of tests with a steam pressure of 5 pounds gage and a water velocity of about four feet per second.

TABLE I.—TEMPERATURE GRADIENT IN A ¾-INCH COPPER TUBE.

Outside tube diameter, inch	.750
Inside tube diameter, inch	.655
Length, feet	4.0
Water-heating surface, square foot	.6853
Steam-heating surface, square foot	.7873

No.	Steam temperature, degrees F.	Water temperature, degrees F.		Average temperature difference.	Water heated per hour, pounds.	Heat transfer B.t.u. per hour per sq. foot steam surface.
		Inlet.	Outlet.			
1	2	3	4	5	6	7
1	228.4	55.6	93.4	143.9	2,070	99,060
2	228.0	93.2	125.2	118.8	2,095	85,039
3	228.4	125.1	152.3	89.7	2,106	72,771
4	228.1	152.5	173.4	65.1	2,120	56,007
5	227.3	174.4	189.8	45.2	2,106	41,453

The gradient corresponding to columns 2, 3 and 4 is plotted in figure 4. Although the temperature rise per pass varies from 15.4 to 37.8, the average arithmetical temperature difference may be very closely considered to be the mean temperature difference per pass, since the curvature of the temperature gradient is very slight and a straight line connecting any two points varies only slightly from the best smooth curve through the points.

The relation between temperature difference and heat transfer per hour, columns 5 and 7 of Table I, is plotted on logarith-

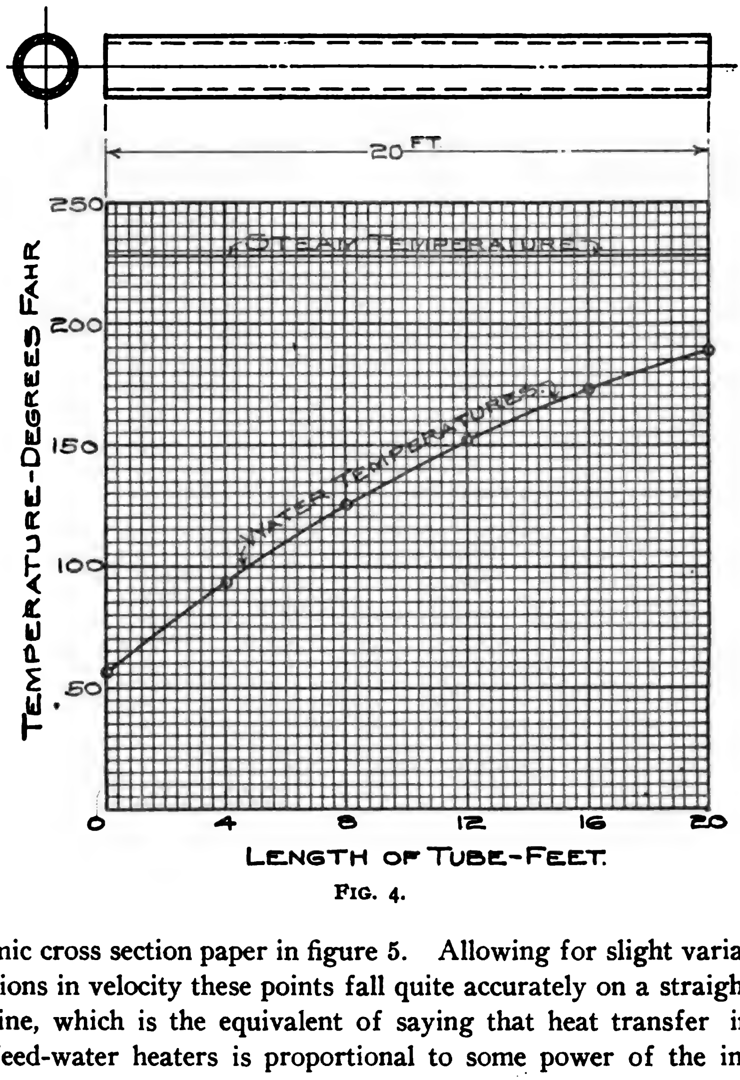
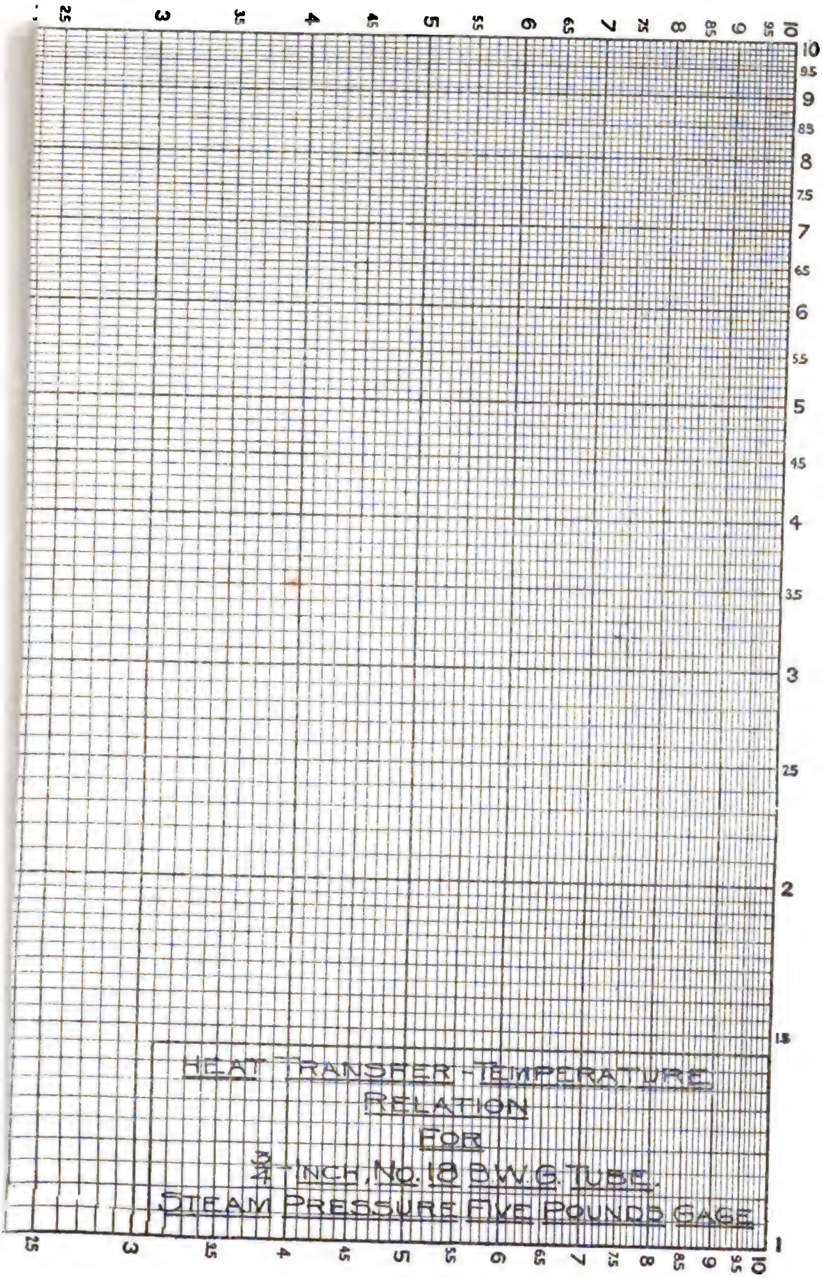


FIG. 4.

mic cross section paper in figure 5. Allowing for slight variations in velocity these points fall quite accurately on a straight line, which is the equivalent of saying that heat transfer in feed-water heaters is proportional to some power of the instantaneous temperature difference. Using the same symbols as before, the result may be expressed analytically as:

$$Q = KS(t_s - t)^n,$$

where n is the slope of the line on the logarithmic curve, in this case a value somewhat less than 1. Referring to figure 3 and


HEAT TRANSFER - TEMPERATURE
RELATION
FOR
3/4 INCH No. 18 B.W.G. TUBE.
STEAM PRESSURE FIVE POUNDS GAGE

proceeding as in the previous case, the heat added in an element of length is:

$$dQ = cWdt = KS(t_s - t)^n \, dl,$$

$$\frac{dt}{(t_s - t)^n} = \frac{KSdl}{cW}.$$

Integrating:

$$\frac{1}{n-1}(t_s - t)^{1-n} = \frac{KS}{cW} l + \text{constant}.$$

Substituting between limits of t_i and t_o for a length of tube l',

$$\frac{1}{1-n}\left[(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}\right] = \frac{KS}{cW} l',$$

$$Q = KSl'(t_m)^n = cW(t_o - t_i),$$

$$\frac{KSl'}{cW} = \frac{t_o - t_i}{(t_m)^n} = \frac{1}{1-n}\left[(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}\right].$$

Hence,

$$t_m = \frac{(1-n)(t_o - t_i)}{(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}}.$$

This formula is the same as that derived by Mr. Orrok for condensers, but is obtained from the basic experimental proof that heat transfer is proportional to a power of the temperature difference instead of the secondary fact that rate of heat transmission per degree temperature difference is proportional to a power of temperature difference. The latter method is somewhat more involved, inasmuch as it introduces another variable factor, U, which varies with temperature. The relation of U as given in the second column of Table I, page 1152 of Mr. Orrok's paper* may be omitted, as it is without significance in the theory. The important thing is the value of K, column 1 of Mr. Orrok's table, and this is tacitly admitted in the curves of temperature rise, page 1153, where there are presented curves of variable K and N.

*A. S. M. E., 1910.

Furthermore, there is a decided disadvantage in obtaining U as function of t, because the whole purpose is to obtain an experimental value of K *which will remain constant throughout* the heater design in question.

The value of U in Mr. Orrok's equation is the coefficient which would have to be inserted in order that the correct basic form,

$$Q = K(t_s - t)^n S,$$

may be converted to and have the same numerical value as

$$Q = U(t_s - t)\, S.$$

Hence

$$\frac{U}{K} = (t_s - t)^{1-n},$$

$$U = K(t_s - t)^{1-n}.$$

This makes U, the coefficient of heat transmission, a function of the temperature difference and introduces an undesirable complication, because the natural thing to do is to search for a coefficient which will remain constant for a given film agitation. Naturally for fractional values of n the value of K in formula on page 267 will be higher than the value derived from the logarithmic mean, the amount of variation being greater for departures of the exponent, n, from unity.

A relation between the coefficients, K and K′, from the two equations may be established as follows:

$$\frac{KSl'}{cW} = \frac{1}{1-n}\left[(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}\right],$$

$$\frac{K'Sl'}{cW} = \log_e \frac{(t_s - t_i)}{(t_s - t_o)},$$

$$\frac{K}{K'} = \frac{(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}}{(1 - n)\log_e \dfrac{(t_s - t_i)}{(t_s - t_o)}},$$

$$= \frac{(t_s - t_i)^{1-n} - (t_s - t_o)^{1-n}}{\log_e \left(\dfrac{t_s - t_i}{t_s - t_o}\right)^{1-n}}.$$

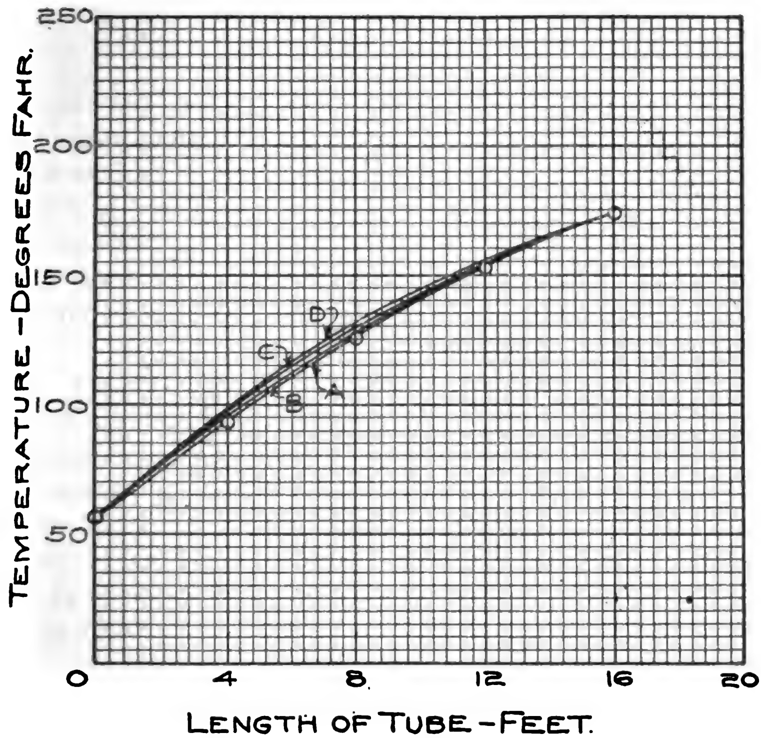


FIG. 6.

A comparison of the two laws of temperature variation for the range of data in Table I is shown in the curves of figure 6. The lines A, B and C are the gradients from the relation

$$l = c\,(t_s - t)^n$$

when the values of n are 0.7, 0.8 and 0.9. The line D is the gradient from the relation

$$l = C \log_e (t_s - t)$$

and the small circles mark the inlet and outlet points in columns 2 and 3 of Table I.

IV.—Heat Transfer in Marine Heaters. Bureau S. E. Heater.

Having established the general law for heat transfer, the next step is to apply the law to the test of heaters used in the Naval Service. Two entirely different types of heaters which have been exhaustively investigated are (1) a feed heater designed by the Bureau of Steam Engineering for battleships 34 and 35 and extensively used in evaporator plants on board ship; and (2) the spirally corrugated film heater manufactured by Schutte and Koerting.

The Bureau feed-water heater is illustrated in Plate I. The apparatus tested consists of a composition shell containing 117 semi-circular ¾-inch tubes No. 16 B.W.G., expanded into a composition tube sheet, and a cast-steel bonnet, the bonnet being cast to form separate passages over the two ends of the tubes. When used for feed heating the feed water passes through the tubes and the steam circulates in the outer casing. In the heater tested the tubes vary in length from 20.64 inches on the inner row of 5⅜-inch radius to 69.72 inches on the outer row of 21-inch radius, the mean tube length being 45 inches from outside face of tube sheet. The total heating surface is 88.2 square feet, of which 86 square feet is tube area. The net area through the water passage of tubes is .362 square foot.

The original tests of this heater, conducted by Mr. O. Z. Howard, M. E., were reported in the Journal of the American Society of Naval Engineers, February, 1912, pp. 155–166.

The method of test and the test apparatus is described in detail in the reference above and there is reprinted here, as Table II, the data and results of forty one-hour tests. The tests are divided into two main groups dependent on the inlet temperature, which was maintained at about 80 degrees F. in the first case and from 130 to 150 degrees in the second. Further subdivision was according to feed velocity and steam pressure. The mean water velocities chosen were 35, 71, 107,

143 and 171 feet per minute, corresponding to deliveries of about 48,000, 96,000, 143,000, 190,000 and 230,000 pounds of water per hour or 410, 820, 1,220, 1,622 and 1,960 pounds of water per tube per hour. The steam pressures were 5, 10, 15 and 20 pounds gage and the water pressure was held at from 230 to 250 pounds gage. The tests show in general an increase of heat transfer with steam pressure and with velocity, but a decrease at the same pressure with increasing temperature difference. These tests do not form as good material for analysis as those which followed; furthermore, some doubt is cast due to irregularities in data caused, as was subsequently determined, from the accumulation of air in the steam space.

Table III gives the data from additional tests which were made at a constant water velocity of 130 feet per minute, constant steam pressure of 10 pounds gage and variable temperature of inlet water. The first test was run with an inlet temperature of 75 degrees, the outlet temperature from this test being used as the inlet temperature of the second test, and so on, with the inlet of each succeeding test corresponding closely to the outlet of the previous run until an outlet temperature of 208 degrees F. was reached.

The tubes were then fitted with retarders consisting of annealed copper strips, 5/8 inch wide and 0.0268 inch thick, twisted into a spiral of six inches pitch, after which the series of tests was repeated. The drop in pressure through each pass of the heater as measured by differential mercury gage was 0.55 inch of mercury without retarders as against 0.98 with retarders, hence the introduction of the retarders involved no serious increase in friction. Subsequently the tests were extended under the writer's direction to water rates of 2,025, 2,500 and 3,030 pounds of water per tube. The inlet water temperature was taken in the water inlet nozzle and the outlet at two points in a lagged discharge pipe far enough beyond the outlet nozzle to insure the thorough mixing of the water by a spiral deflector or mixer. The water was measured by a 6-inch × 3-inch Venturi meter, which indicated according to

TABLE II.—RESULTS OF

Number of test.	Date.	Duration of test (hours).	Barometer, corrected.	Room temperature, degrees F.	Water side. River water pumped through heater, pounds per hour.	Equivalent fresh water pumped through heater, pounds per hour.	Temperature at inlet, degrees F.	Temperature at outlet, degrees F.	Degrees rise.	B.T.U.s absorbed by water, per hour.	Pressure at pump, pounds gage.	Steam side. Steam condensed, pounds per hour.	Pressure at inlet, inches mercury.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
	May.												
1	6	1	29.66	79.0	49248	47574	84.2	121.8	37.6	1788782	281.7	1917.5	9.08
2	6	1	29.66	79.0	48721	47064	86.3	124.7	38.4	1807258	226.6	1957.5	19.59
3	6	1	29.66	78.0	49690	48001	88.7	128.9	40.2	1929640	236.0	2100.0	29.58
4	6	1	29.59	78.0	49630	47943	90.8	136.7	45.9	2200584	228.9	2416.0	38.34
5	8	1	29.44	71.0	99580	96194	82.8	112.9	30.1	2895439	241.0	3054.5	9.06
6	8	1	29.39	71.0	99550	96165	82.7	113.8	31.1	2990732	238.4	3185.0	19.24
7	8	1	29.30	75.0	99003	95637	83.9	119.3	35.4	3385550	245.3	3624.5	29.08
8	8	1	29.33	73.0	98943	95579	87.1	126.3	39.2	3746697	249.6	4029.0	38.60
9	9	1	29.17	76.0	149625	144538	77.1	97.6	20.5	2963029	236.6	3121.0	9.41
10	9	1	29.14	76.5	149550	144465	77.9	101.7	23.8	3438267	239.3	3635.5	19.06
11	9	1	29.10	76.0	149943	144845	81.1	107.7	26.6	3852877	244.3	4108.0	29.34
12	9	1	29.08	75.0	148872	143810	81.1	114.3	33.2	4774492	258.6	5101.0	38.70
13	10	1	29.24	83.0	200357	193545	79.1	102.2	23.1	4470890	251.9	4605.0	9.00
14	10	1	29.22	83.0	199380	192601	77.2	103.2	26.0	5007626	254.0	5211.0	19.08
15	10	1	29.22	83.0	198583	191831	78.7	108.2	29.5	5659015	257.0	5935.0	29.27
16	11	1	29.31	85.5	199160	192389	79.9	112.3	32.4	6233404	252.6	6590.0	38.30
17	11	1	29.30	85.0	239352	231214	79.7	100.2	20.5	4739887	245.7	4932.0	9.43
21	24	1	29.47	75.5	239420	231280	79.1	103.0	23.9	5527592	234.3	5781.0	19.05
22	24	1	29.46	77.0	239462	231320	80.2	107.4	27.2	6291904	225.7	6602.0	29.24
23	24	1	29.41	73.0	238944	230820	81.7	112.2	30.5	7040010	222.1	7434.0	39.27
18	12	1	29.38	85.0	49385	47706	147.2	171.4	24.2	1154485	249.1	1260.0	8.86
19	12	1	29.37	85.0	49390	47711	144.3	170.2	25.9	1235715	259.1	1362.0	19.09
20	12	1	29.31	85.0	49350	47672	143.7	174.1	30.4	1449229	248.9	1599.0	29.28
	June.												
36	1	1	29.09	82.5	49328	47651	151.8	185.0	33.2	1582013	234.1	1756.0	39.25
	May.												
28	29	1	29.46	86.0	99969	96571	139.2	156.2	17.0	1641707	244.1	1769.0	10.48
29	29	1	29.45	87.0	99495	96112	134.7	154.9	20.2	1941462	241.1	2097.0	20.00
30	29	1	29.44	87.0	99277	95902	133.3	155.6	22.3	2138615	255.0	2329.0	30.71
31	29	1	29.44	87.0	98930	95566	135.6	160.4	24.8	2370037	254.3	2592.0	39.45
32	31	1	29.36	82.0	147376	142365	145.6	161.1	15.5	2206658	253.9	2330.0	9.39
33	31	1	29.36	80.0	146867	141874	144.3	162.5	18.2	2582107	246.0	2754.0	19.48
34	31	1	29.23	81.0	146763	141773	146.4	166.2	19.8	2807105	240.4	3015.0	29.05
35	31	1	29.26	81.0	147631	142612	148.4	170.6	22.2	3165986	247.9	3415.0	39.37
	June.												
37	1	1	29.12	84.0	195842	189183	149.8	163.1	13.3	2516134	249.4	2652.0	9.61
38	1	1	29.12	81.5	197820	191094	147.8	161.6	13.8	2637097	252.4	2814.0	19.50
39	1	1	29.15	81.5	195743	189088	148.1	163.6	15.5	2930864	237.6	3123.0	29.66
40	1	1	29.15	81.5	195604	188953	150.4	168.6	18.2	3438945	242.0	3690.0	38.89
	May.												
24	25	1	29.35	78.5	237447	229374	128.3	132.3	4.0	917496	244.4	990.0	9.89
25	25	1	29.35	78.5	237632	229553	131.0	144.8	13.8	3167831	248.4	3362.0	19.90
26	25	1	29.31	79.8	238008	229916	132.2	148.1	15.9	3655664	250.7	3899.0	30.00
27	25	1	29.30	81.0	237546	229460	129.8	147.0	17.2	3946712	258.7	4213.0	38.86

Steam side.														
Pressure at inlet, pounds absolute.	Temperature at inlet, degrees F.	Temperature at inlet corresponding to pressure, degrees F.	Degrees superheat.	Temperature of outlet steam water to weighing tanks, degrees F.	B.T.U.s rejected by steam.	Radiation from heater, B.T.U.s per hour.	Radiation from heater, calculated from test, B.T.U.s per hour.	Equivalent dry saturated steam condensed, pounds per hour.	Velocity of water through tubes, feet per minute.	Heat transfer per sq. ft. heat'g surf. per deg. differ. (temp. due to pressure, column 17).	Mean temperature difference, geometrical (temperature due to pressure, column 17).	Heat transfer per sq. ft. heat'g surf. per deg. mean temp. diff. (Temp. due to pressure.)		
15	16	17	18	19	20	21	22	23	24	25	26	27		
19.03	227.1	226.0	1.1	218.4	1858878	70096	71088	1934	35.3	148.6	122.3	165.8		Temperature difference over 122 degrees.
24.20	239.3	238.3	1.0	230.1	1882880	75622	76944	1975	35.0	140.4	131.2	156.1	*a*	
29.09	249.2	248.6	0.6	239.6	2006747	77107	82176	2121	35.7	142.3	137.3	159.3		
33.38	257.6	256.8	0.8	251.7	2285758	85174	86208	2430	35.7	156.1	142.5	175.0		
18.96	232.5	224.9	7.6	216.9	2974313	78874	77520	3091	71.4	237.2	125.9	260.7		
23.89	240.9	237.5	3.4	227.7	3073945	83213	81552	3223	71.4	225.1	139.4	243.2		
28.67	250.9	247.7	3.2	238.6	3470705	85155	84432	3665	71.0	240.2	143.4	267.6	*b*	
33.20	262.8	256.2	6.6	248.3	3836551	89854	91104	4077	71.1	257.2	149.4	284.3		
18.94	231.9	225.0	6.9	214.8	3044573	81544	74832	3165	107.1	233.3	138.1	243.2		
23.66	241.3	237.0	4.3	226.2	3515398	77131	79104	3685	107.1	250.4	143.8	271.0		
28.70	250.9	247.8	3.1	238.3	3934720	81843	83952	4156	107.4	267.5	152.9	285.6	*c*	
33.28	259.5	256.3	3.2	245.2	4864660	90168	88560	5170	106.7	314.7	160.4	337.4		
18.78	249.3	224.6	24.7	211.7	4544932	74042	79824	4723	143.5	354.1	132.8	381.6		
23.72	258.1	237.2	20.9	224.0	5092356	84730	84048	5338	142.8	360.7	149.4	379.9		
28.72	253.9	247.8	6.1	231.2	5735519	76504	82032	6058	142.3	384.4	154.8	414.4	*d*	
33.21	264.5	256.2	8.3	242.8	6316957	83553	85920	6712	142.8	406.1	156.5	451.5		
19.02	235.3	225.3	10.0	212.9	4828428	88541	72144	5020	171.3	375.9	138.1	389.1		
23.83	239.8	237.4	2.4	224.0	5598043	70451	78864	5869	171.3	400.8	144.4	433.9		
28.83	258.3	248.1	10.2	233.3	6380160	88256	87024	6740	171.6	420.7	156.3	456.3	*e*	
33.63	269.6	256.9	12.7	242.8	7143502	103492	94368	7595	171.3	462.1	160.0	498.6		
18.78	225.4	224.6	0.8	218.9	1220038	65553	67392	1268	36.0	178.7	65.1	201.0		Temperature difference below 102 degrees.
23.80	238.1	236.3	1.8	229.6	1310467	74752	73488	1373	36.0	161.5	78.7	178.0		
28.77	248.9	247.9	1.0	241.1	1525590	76361	78672	1611	36.0	165.9	88.5	185.6	*f*	
33.56	258.4	256.8	1.6	250.3	1664484	82471	84432	1770	36.1	179.7	87.7	204.5		
19.62	229.9	226.9	3.0	218.9	1716355	74648	69072	1786	72.6	221.8	79.0	235.6		
24.29	242.2	238.5	3.7	232.4	2015167	73705	74496	2114	72.2	220.0	93.9	234.4		
29.54	253.3	249.4	2.9	243.7	2219120	80505	79344	2347	72.0	216.6	103.7	233.8		
33.84	258.4	257.3	1.1	251.5	2453688	83651	82272	2609	71.8	228.5	107.3	250.4		
19.04	238.1	225.3	12.8	216.6	2275581	68923	74928	2366	107.2	323.6	72.1	346.9		
23.99	247.9	237.8	10.1	228.3	2665453	83346	80592	2795	106.9	323.1	84.6	346.0		
29.62	250.0	249.6	0.4	238.8	2884137	77032	81120	3050	106.9	316.8	92.1	345.5		
33.71	259.9	257.1	2.8	248.2	3246545	80559	85872	3452	107.6	338.5	96.1	373.4		
19.02	230.5	225.3	5.2	216.9	2579590	63456	70320	2682	142.6	387.3	69.8	408.6		
23.88	240.1	237.5	2.6	225.0	2722680	85583	76128	2855	144.0	344.0	83.4	358.4		
28.89	262.0	248.2	13.8	233.9	3021696	90832	86640	3192	142.5	342.2	93.7	354.6		
33.47	257.5	256.6	0.9	243.5	3521518	82573	84480	3743	142.6	375.8	95.5	408.2		
19.28	226.5	226.0	0.5	180.3	997168	79672	71040	1037	171.6	115.7	102.0	102.0		
24.19	239.3	238.2	1.1	223.9	3254846	87015	77184	3414	172.0	344.1	98.7	363.8		
29.13	250.1	248.6	1.5	235.1	3745126	89462	81744	3958	172.4	364.6	107.1	386.9		
33.47	257.8	256.6	1.2	241.8	4028420	81708	84864	4282	171.9	360.1	123.0	363.7		

TABLE III.—TEST OF

(With Velocity of 131 feet per minute,

					Water side.							Steam side.			
Test number.	Date, September, 1911.	Duration of test, hours.	Barometer, corrected.	Room temperature, degrees F.	River water pumped through heater, pounds per hour.	Equivalent fresh water pumped through heater, pounds per hour.	Temperature at inlet, degrees F.	Temperature at outlet, degrees F.	Degrees rise.	B.T.U.s absorbed by water, per hour.	Pressure at pump, pounds gage.	Steam condensed, pounds per hour.	Pressure at inlet, inches mercury.	Pressure at inlet, pounds absolute.	Temperature at inlet, degrees F.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
															WITH
1	20	1	29.97	76.0	178543	172473	75.0	108.2	33.2	5726104	55	5943	19.46	24.28	252.6
2	20	1	29.96	79.0	179933	173815	108.9	135.6	26.7	4640861	50	4870	19.48	24.29	252.2
3	21	1	30.13	74.5	176311	170316	136.3	158.8	22.5	3832110	50	3978	20.10	24.67	267.5
4	21	1	30.11	75.0	178260	172199	158.8	176.5	17.7	3047922	50	3208	19.52	24.37	255.2
5	21	1	30.09	75.0	178357	172293	175.6	189.6	14.0	2412102	50	2665	19.70	24.46	244.5
6	21	1	30.07	77.0	175999	170015	189.5	200.8	11.3	1921170	50	2278	19.80	24.50	262.6
7	22	1	30.09	78.5	175731	169756	200.1	208.1	8.0	1358048	50	1802	19.95	24.45	243.4
															WITHOUT
8	26	1	30.03	81.0	178041	171988	78.2	105.6	27.4	4712471	50	4997	19.29	24.23	246.7
9	26	1	30.03	80.0	175948	169966	104.3	128.8	24.5	4164167	55	4523	19.23	24.20	244.4
10	27	1	30.17	77.0	178356	172292	130.4	149.9	19.5	3359694	50	3744	19.47	24.37	243.9
11	27	1	30.16	71.0	176922	170907	149 9	166.9	17.0	2905419	53	3237	19.41	24.34	243.6
12	27	1	30.08	73.0	175459	169493	167.9	182.3	14.4	2440699	50	2753	19.42	24.31	242.0
13	27	1	30.07	74.0	173111	167225	182.7	194.1	11.4	1906365	55	2321	19 30	24.24	242.2
14	28	1	30.11	74.0	171906	166061	192.8	200.5	7.7	1278670	55	1748	19.20	24.21	245.2
15	28	1	30.10	75.0	171695	165857	200.5	206.8	6.3	1044899	50	1552	19.36	24.29	240.5

FEED-WATER HEATER.

and steam pressure of 25 pounds absolute.)

Steam side.											
Temperature at inlet, corresponding to pressure, degrees F.	Degrees superheat.	Temperature of outlet steam water to weighing tank, degrees F.	B.t.u.s rejected by steam.	Equivalent dry saturated steam condensed, pounds per hour.	B.t.u.s unaccounted for.	Component velocity of water through tubes (parallel to axis) feet per minute.	Heat transfer per hour per sq. ft. of heating surface, per degree difference (temp. due to press., col. 17).	Mean temperature difference, geometrical (temp. due to press., col. 17).	Heat transfer per hour per sq. ft. heating surface, per degree mean temperature difference (temp. due to press., col. 17).	Pounds of dry steam condensed per sq. ft. of heating surface.	27 reduced to condensation per sq. ft. of tube surface (86 square feet).
17	18	19	20	21	22	23	24	25	26	27	28
RETARDERS.											
238.44	14.16	231.7	5744979	6028	18875	135.7	397.1	146.19	443.99	68.33	69.33
238.47	13.73	232.0	4705297	4937	64436	137.4	406.0	116.34	452.17	55.96	56.78
239.34	28.16	232.4	3870674	4063	38564	135.4	421.6	91.43	475.10	46.06	46.75
238.65	16.55	231.8	3104702	3258	56780	137.7	432.7	70.80	487.98	36.93	37.49
238.86	5.64	232.1	2564653	2691	152561	138.4	432.2	56.00	488.24	30.50	30.94
238.95	23.65	233.6	2208589	2318	287419	136.9	431.7	43.58	499.70	26.27	26.56
239.13	4.27	232.5	1729902	1816	371854	137.4	394.4	34.98	440.07	20.58	20.88
RETARDERS.											
238.33	8.37	224.4	4852641	5091	140170	130.5	333.6	146.29	365.14	57.71	...
238.26	6.14	229.9	4362660	4577	198493	130.0	355.0	121.70	387.86	51.88	...
238.65	5.25	232.1	3602028	3780	242334	131.8	351.8	98.09	388.24	42.85	...
238.58	5.02	232.6	3112181	3265	206762	131.4	371.4	79.92	412.08	37.01	...
338.51	3.49	233.8	2641448	2771	200749	131.0	391.8	63.19	437.82	31.41	...
238.35	3.85	233.9	2226976	2336	320611	129.8	388.3	49.67	435.06	26.48	...
238.28	6.92	333.0	1681279	1764	402609	129.2	318.7	41.49	349.34	19.99	...
238.47	2.03	233.3	1488772	1562	443873	129.4	311.9	34.71	341.23	17.71	...

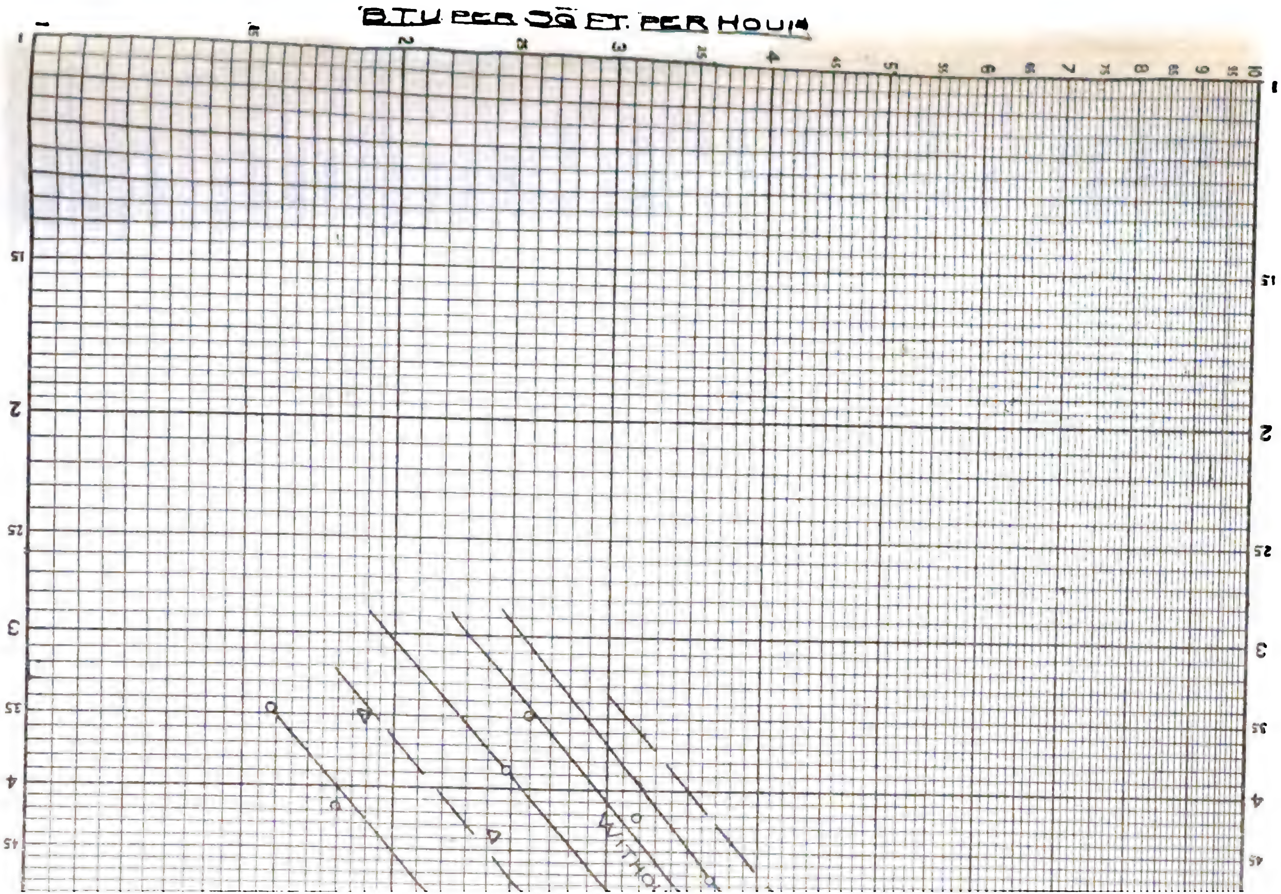
calibrations before and after tests 1.5 per cent. low at all rates.

Steam was delivered from a low-pressure steam line to the 5-inch steam supply where its temperature and pressure were taken. There was always some superheat, hence a calorimeter was not needed. Steam pressure was again taken at the bottom of the steam space. Air cocks were fitted at both top and bottom of the steam space and allowed to blow continuously. Only in this way could consistent results be obtained, as considerable air accumulated in the lower portion of the steam space and reduced heating effect if allowed to remain. In a subsequent section the effect of this air in steam will be treated in greater detail.

The results of the second series of tests are given in Table IV for plain tubes and in Table V for tubes with retarders. It should be noted that in several tests the steam pressure at inlet, column 15, is higher than the ten pounds fixed on for the test. This was necessary because at low water temperatures and high rates of flow the steam pressure drop from top to bottom of heater, as shown by a comparison of columns 15 and 17, was so great that with 10 pounds on the shell the outlet pressure was not sufficient to deliver the condensate through the cooling coil to the weighing tanks and maintain a water-free condition throughout the heating surface. However, in almost all of these cases the average steam temperatures, column 22, corresponding to the two pressures was practically 238 degrees, the saturation temperature of 10 pounds gage. The pressure drop within the steam space is further indicated by column 24, temperature of condensed steam, which is far below the temperature of saturation, column 20, corresponding to inlet pressure and even lower than that of column 22, corresponding to outlet pressure.

The data from these tests have been plotted on logarithmic cross-section paper in figure 7, where the abscissae are temperature differences in degrees F., and the ordinates are heat transfer in B.t.u. per hour per square foot of heating surface.

There can be no doubt that for the heater in question the heat transfer—temperature relation is given by:


B.T.U. PER SQ. FT. PER HOUR
WITHO

					Water Data.								
1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	31	30	29.93	62	236899	73.3	105.1	29.8	.968	6033979	9.5	0.39	0.18
2	31	30	29.90	64	236913	106.3	131.4	25.1	.958	5697758	9.5	0.40	0.18
3	31	35	29.88	66	237400	129.2	150.2	21.0	.952	4740698	9.7	0.45	0.20
4	31	30	29.82	69	238980	150.5	168.3	17.8	.945	4007992	9.6	0.40	0.18
5	31	30	29.75	71	237624	168.2	182.5	14.3	.940	3093392	9.6	0.40	0.18
6	31	30	29.74	71	239140	182.8	194.8	12.0	.937	2687734	9.0	0.40	0.18
7	31	30	29.71	72	238452	194.8	204.6	9.8	.933	2179451	7.9	0.46	0.21
8	1	30	29.95	59	236909	75.0	102.0	27.0	.968	7745517	11.2	0.98	0.45
9	1	30	29.99	59	236090	102.7	125.6	22.9	.962	6526941	11.2	0.87	0.40
10	1	30	30.04	59	237305	124.8	143.0	18.2	1.040	5647984	10.9	1.14	0.52
11	1	30	30.04	60	236646	142.4	158.3	15.9	1.032	4867961	10.6	1.12	0.51
12	1	30	30.06	61	237476	157.4	171.4	14.0	1.026	4268803	10.7	1.20	0.55
13	1	35	30.11	60	236927	170.9	183.1	12.2	1.022	3699526	10.3	1.14	0.52
14	1	30	30.15	60	237390	182.0	192.6	10.6	1.020	3214786	11.1	1.21	0.55
15	1	30	30.19	60	236933	192.1	202.4	9.3	1.016	2806206	11.2	1.26	0.57
16	3	30	30.09	54	356864	73.9	97.9	22.4	1.042	8320618	13.1	1.78	0.81
17	3	30	29.98	56	356799	98.1	119.1	21.0	1.037	7770080	13.1	1.80	0.82
18	3	30	29.91	58	359454	119.3	135.9	16.6	1.030	6197413	13.5	1.83	0.83
19	3	30	29.83	60	357954	136.1	151.2	15.1	1.029	5560605	13.7	1.76	0.80
20	3	30	29.81	61	358776	150.9	163.8	12.9	1.027	4754432	13.5	1.80	0.82
21	3	30	29.78	62	358488	164.0	175.5	11.5	1.022	4212734	13.6	1.87	0.85
22	3	30	29.76	69	358855	175.4	186.0	10.6	1.021	3884811	13.4	1.97	0.90
23	3	30	29.72	65	359856	186.3	195.1	8.8	1.020	3231507	11.6	1.90	0.87
24	4	30	30.04	59	356986	194.9	201.9	7.0	1.018	2543320	10.8	1.73	0.79
25	4	30	30.04	59	358480	202.0	208.0	7.0	.930	2283356	11.2	1.69	0.77

	Steam Data.															
1	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
1	10.76	25.47	4.36	19.07	273.6	241.1	32.5	225.4	233.3	218.1	3356.0	6756	990.0	6708040	-103939	6884
2	9.76	24.45	3.63	20.32	263.7	238.8	24.9	228.8	233.8	222.5	2957.5	5875	981.2	5764550	66792	5734
3	9.89	24.57	7.39	22.67	257.7	239.1	18.6	233.3	236.2	226.6	2883.0	4940	994.1	4814000	73324	4902
4	9.99	24.64	6.37	23.02	255.4	239.3	16.1	235.5	237.4	232.0	2245.0	4050	969.6	4152004	143492	4380
5	9.88	24.50	8.84	23.46	240.8	239.0	1.8	236.6	237.8	233.7	2714.5	3429	999.0	3288411	94873	3437
6	9.99	24.60	9.38	23.99	241.3	239.2	2.1	237.8	238.5	235.3	1434.5	2869	997.6	2747334	59400	2872
7	9.96	24.56	9.68	24.18	239.6	239.1	0.5	238.4	238.8	236.4	1180.0	2360	995.6	2259826	79765	2360
8	13.40	28.32	5.81	20.39	280.0	247.0	33.0	229.4	238.2	216.8	3968.5	7937	993.8	7887792	142274	8048
9	11.03	25.77	4.96	19.70	278.6	241.7	36.9	227.2	234.5	220.9	3388.0	6776	989.6	6705530	180589	6878
10	10.11	24.87	5.72	20.48	268.2	239.8	28.4	229.2	234.5	225.0	2799.0	5598	980.7	5489239	254255	6490
11	10.08	24.84	7.19	21.95	265.4	239.7	25.7	233.0	236.4	229.1	2672.5	5345	975.3	5212979	345028	5404
12	10.15	24.92	8.24	23.00	262.7	239.9	22.8	233.5	237.7	232.0	2336.5	4673	971.1	4537950	169067	4725
13	10.13	24.92	8.81	23.60	261.9	239.9	22.0	236.9	238.4	234.5	2340.0	4011	968.2	3883490	183864	4047
14	9.94	24.73	8.92	23.73	255.1	239.5	15.6	237.2	238.4	235.3	1733.5	3467	963.2	3339414	124628	3488
15	9.98	24.81	9.39	24.22	244.6	239.7	4.9	238.3	239.0	236.5	1448.5	2897	957.8	2774747	-52459	2903
16	15.06	29.81	6.39	21.14	290.4	249.9	40.5	251.0	240.5	227.9	4230.5	8461	997.3	8438199	116687	8605
17	13.71	28.44	6.11	20.84	288.4	247.3	41.1	230.2	238.8	221.0	3924.0	7848	993.5	7796988	26908	7981
18	9.98	24.68	4.39	19.09	280.0	239.4	40.6	225.2	232.9	219.7	3178.0	6356	991.8	6303881	146468	6464
19	10.01	24.67	5.63	20.29	274.7	239.3	35.4	228.8	234.1	224.6	2901.5	5803	984.3	5711893	149288	5890
20	10.02	24.67	7.23	21.88	268.2	239.3	28.9	232.8	236.1	227.9	2543.5	5087	977.8	4974069	227637	5148
21	9.96	24.59	8.04	22.67	256.8	239.2	17.6	234.7	237.0	229.6	2207.0	4414	970.7	4284670	71836	4445
22	10.15	24.77	8.71	23.33	259.1	239.6	19.5	236.3	238.0	234.0	2049.0	4098	967.3	3963999	81184	4132
23	10.04	24.64	9.18	23.78	250.2	239.3	10.9	237.3	238.3	234.5	1716.5	3433	962.6	3304606	73099	3447
24	10.02	24.76	9.89	24.61	246.2	239.6	6.6	239.2	239.4	235.0	1338.0	2676	960.1	2569288	25928	2684
25	10.01	24.77	9.88	24.64	243.7	239.6	4.1	239.3	239.5	236.8	1227.0	2454	957.1	2348783	65387	2458

	Deduced Data.							
1	31	32	33	34	35	36	37	38
1	175	3085	77463	143.1	142.9	541.3	542.1	27.1
2	176	3085	64585	114.9	114.6	560.1	563.6	24.2
3	177	3089	55738	96.5	96.2	556.9	558.6	22.0
4	179	3037	45409	78.0	77.6	580.7	585.7	18.1
5	179	3058	36398	62.4	62.1	580.1	582.9	14.4
6	181	3044	30467	49.7	49.4	613.0	626.7	12.0
7	181	3098	24705	39.1	38.8	651.8	656.7	9.9
8	229	3533	87795	149.7	149.5	586.5	587.3	27.2
9	229	3532	73869	120.3	120.2	624.0	624.6	23.2
10	222	3542	65993	100.6	100.3	634.1	636.0	20.4
11	222	3535	55279	86.0	85.6	641.6	644.6	18.2
12	223	3541	48388	73.3	73.2	660.1	661.0	15.9
13	223	3538	41935	61.4	61.1	683.0	686.3	13.6
14	224	3542	36440	51.1	51.0	713.1	714.5	11.7
15	224	3538	31608	42.2	42.1	723.7	725.5	9.8
16	263	3050	94332	153.8	153.6	613.3	614.1	24.1
17	264	3049	88074	130.2	130.1	676.5	677.0	22.4
18	267	3072	69794	104.7	104.6	666.6	667.8	18.0
19	267	3059	63052	90.4	90.4	697.5	697.5	16.2
20	269	3068	55714	78.7	78.8	683.1	684.2	14.9
21	269	3064	47745	67.2	67.1	710.6	711.6	12.4
22	270	3067	44012	57.3	57.1	768.1	770.8	11.5
23	271	3075	36689	47.6	47.4	769.6	772.8	9.6
24	269	3052	28852	41.0	40.8	703.8	707.1	7.5
25	266	3000	25904	35.0	34.9	740.2	742.3	7.0

*U. S. Naval Academy.

TABLE IV.

TEST OF BUREAU HEATER WITH RETARDERS.

Columns 1–5; Water Data, columns 6–14.

1	2	3	4	5	6	7	8	9	10	11	12	13	14
26	1	30	29.62	74	235198	65.1	98.0	32.9	.950	7351039	11.7	1.00	0.46
27	1	30	29.62	74	235906	98.3	124.4	26.1	.944	5812724	10.1	0.82	0.37
28	2	30	29.90	64	235658	128.5	151.6	23.1	.937	5110988	11.6	0.78	0.35
29	2	30	29.90	64	235077	151.9	170.7	18.8	.936	4257355	11.8	0.71	0.32
30	2	30	29.89	71	235901	170.7	185.3	14.6	.932	3210613	11.9	0.80	0.36
31	2	30	29.89	71	236890	185.0	196.6	11.6	.930	2549569	11.7	0.80	0.36
32	2	30	29.85	72	236955	197.4	206.6	9.2	.928	2087996	11.6	0.80	0.36
33	3	30	29.80	67	290432	75.1	108.1	33.0	.947	9076000	8.5	1.30	0.50
34	3	30	29.77	72	290009	108.5	136.1	27.6	.942	7540791	9.9	1.07	0.49
35	3	30	29.77	72	290451	136.1	160.3	24.2	.938	6593238	9.4	1.08	0.47
36	3	30	29.75	70	288780	160.2	176.3	16.1	.934	4307309	9.3	1.06	0.48
37	3	30	29.75	70	289011	176.9	192.1	14.2	.932	3823165	9.2	1.02	0.46
38	3	30	29.65	81	284237	190.6	202.8	11.2	.929	2955205	9.1	1.00	0.46
39	4	30	29.66	81	349317	75.5	106.8	31.3	.948	10364235	10.2	1.60	0.73
40	4	30	29.66	81	351686	105.7	132.3	25.6	.943	8488952	10.4	1.60	0.73
41	4	30	29.62	80	350304	132.2	152.2	20.0	.939	6680401	12.9	1.60	0.73
42	4	30	29.62	80	352480	152.0	168.7	17.7	.936	5820024	12.7	1.60	0.73
43	5	30	29.81	70	358800	168.4	182.3	13.9	.933	4675846	11.7	1.67	0.75
44	5	30	29.81	70	350580	182.1	193.5	11.4	.930	3735929	11.6	1.68	0.76

Steam Data, columns 15–27.

1	15	16	17	18	19	20	21	22	23	24	25	26	27
26	12.73	27.28	6.32	20.87	275.4	245.0	30.4	230.3	237.7	216.6	3739.5	7479	990.0
27	9.55	24.20	5.45	20.00	277.1	238.0	39.1	228.0	233.0	216.4	3033.5	6067	993.9
28	9.58	24.27	6.38	21.07	265.3	238.5	26.8	230.8	234.7	215.9	2651.0	5302	978.7
29	9.58	24.27	7.54	22.23	260.5	238.5	22.0	233.7	236.1	219.4	2275.5	4551	972.8
30	9.58	24.27	8.37	23.06	258.0	238.5	19.5	235.6	237.1	231.7	2790.5	5581	969.3
31	9.58	24.27	8.98	23.67	253.2	238.5	14.7	237.1	237.8	234.1	1402.5	2805	964.6
32	9.58	24.25	9.28	23.95	238.1	238.4	0.0	237.7	238.1	236.0	1086.5	2173	997.6
33	17.50	32.24	7.18	21.82	290.2	254.3	35.9	239.7	243.5	224.4	4630.0	9260	990.0
34	12.87	27.50	5.83	20.46	283.2	245.4	37.8	229.2	237.3	223.1	3820.5	7641	989.2
35	13.60	28.23	9.35	23.98	279.2	246.9	32.3	237.8	242.4	234.3	3358.5	6717	975.8
36	9.58	24.20	7.60	22.22	258.1	238.3	19.8	235.6	238.0	230.0	2273.5	4547	971.1
37	9.58	24.20	8.58	23.20	257.0	238.3	18.7	236.0	237.8	232.3	1742.0	3484	968.3
38	9.58	24.15	9.19	23.76	242.8	238.0	4.8	237.2	237.6	233.2	1515.0	3030	958.6
39	22.22	36.79	9.83	24.40	291.1	262.3	28.8	256.7	250.5	229.8	5377.5	10755	983.8
40	16.04	30.61	6.95	21.52	284.4	251.4	33.0	232.9	241.7	223.3	4323.5	8647	988.7
41	11.64	26.19	5.86	20.41	277.9	242.7	35.2	229.0	235.9	222.9	3375.0	6750	987.1
42	11.78	26.33	7.87	22.42	274.9	243.0	31.9	234.1	238.6	230.7	2965.0	5930	977.7
43	9.58	24.03	7.25	21.80	260.8	238.4	22.4	232.6	235.5	228.4	2381.0	4762	974.2
44	9.58	24.03	8.21	22.66	259.8	238.4	21.4	235.2	236.8	232.5	1954.5	3909	970.5

Steam Data, columns 28–30; Reduced Data, columns 31–38.

1	28	29	30	31	32	33	34	35	36	37	38
26	7404210	53271	7592	174	2010	83384	156.1	155.3	533.8	536.5	38.28
27	6029910	217286	6156	175	2016	65887	122.6	120.9	541.8	545.0	26.21
28	5189067	78079	5371	176	2024	57533	94.6	94.3	612.4	614.3	22.79
29	4452639	95298	4539	177	2009	46897	74.8	74.4	627.0	630.3	18.73
30	5305539	189706	5556	178	2016	56358	59.1	58.8	625.8	628.9	14.99
31	2708071	153522	2905	179	2020	28900	47.0	46.6	614.9	620.2	12.30
32	2060865	57069	2173	180	2005	22958	56.1	55.9	655.4	658.9	9.17
33	9169400	91400	9407	225	2482	100876	152.9	152.1	677.2	680.8	32.45
34	7558477	18043	7786	226	2479	83468	115.0	114.3	743.2	747.8	26.85
35	6554448	-38790	6805	228	2482	74734	94.2	93.6	773.4	778.4	23.50
36	4415538	73943	4552	228	2468	49881	65.7	65.4	749.2	752.6	15.92
37	3373057	90552	3525	229	2470	37715	52.2	52.1	712.9	714.3	12.26
38	2904558	-50467	3036	226	2459	33405	42.4	42.2	809.0	813.0	10.68
39	10580769	216534	10906	258	2925	117408	159.4	158.7	737.0	740.3	32.22
40	8549325	42269	8765	262	3005	96425	123.2	122.7	781.0	784.1	24.92
41	6662805	47504	6865	264	3023	75042	94.7	94.3	732.4	735.8	19.48
42	5797762	-46469	6005	264	3004	66005	78.7	78.5	858.8	862.0	17.14
43	4639140	65504	4824	266	3005	52867	60.9	62.1	830.5	835.2	13.64
44	3793685	58656	3932	267	3020	45338	52.2	50.9	846.9	852.8	11.21

TABLE V.

TEST OF BUREAU HEATER WITHOUT RETARDERS.

$$Q = K(t_s - t)^n$$

where

Q = B.t.u. per hour per square foot of H. S.;

K = a constant from experiment;

t_s = average saturation steam temperature within the heater;

t = water temperature;

n = a constant from experiment.

Furthermore n is the slope of the curve on the logarithmic plot and has for this heater a value

$$n = 0.85,$$

which is independent of the velocity and of the use of retarders. The value of K, which is the intersection of the line prolonged to the vertical axis when $(t_s - t) = 1$, is dependent on the velocity and is therefore the function which varies with the more or less efficient removal of the water side film. As illustrated here the use of retarders has the same effect as increasing the velocity by increasing the value of K. The few inconsistencies in points of the second curve are due to the fact that the heater stood for two months between the tests in Table IV and those in Table V and the tubes were probably slightly coated on the water side. It required at least two runs to thoroughly clean the tubes. Before the tests of Table IV the tubes were scoured, cleaned with a wire brush and hot solution of lye-water to put them in best possible condition. This precaution and the care to remove all entrained air will probably account for the higher rates of heat transmission at a velocity of 170 feet per minute than those recorded in Table II.

The steam pressure drops previously noted result from insufficient steam space and could be remedied by wider tube spacing and consequent increase in weight of header and shell. Unless such an increase of steam space were made, abnormally high back pressures for auxiliary machinery supplying the exhaust steam for feed heating would be necessary.

In a bulletin entitled "A Study in Heat Transmission,"* Clement and Garland discuss the condition on the steam side of a heater tube. The tests reported dealt with the heat transfer through the film to the metal wall, and the steam wall temperatures were given by thermo couples soldered into small holes in the surface of the tube. While such a study reveals many points of value it does not arrive at constants useful in design because the metal wall temperature is always an unknown factor. However, the conclusions of Messrs. Clement and Garland in regard to steam side are so much to the point that they are quoted verbatim: "On the steam side of the tube, if a condition of equilibrium is assumed to maintain, there will be the condition of saturated steam imparting its heat to a film of water on the surface of the tube, which film may be assumed to be of a constant thickness, and which is constantly being replaced or renewed by the condensation of fresh steam. The agitation of the steam in contact with the water film will, therefore, if it does not agitate this film, have no effect upon the rate of heat transmission or the conductance, at least, in the case of the present experiments; for the maximum velocity of approach of the steam towards the tube, due to the condensation, is about 1/5 foot per second. This would lead one to believe that the temperature on the surface of separation of the steam from the water film must be practically the temperature of the saturated steam. If such is the case, the agitation of the steam alone would not affect the conductance, as it would not change the temperature drop through the tube. The solution of the problem of more efficient heat transmission from steam to surface of tube, therefore, lies solely in the removal or agitation of the water film."

"It might be well to point out here that it is the rate of agitation and not necessarily the velocity of the water that produces the change in heat transmission. The velocity in feet per second has been used for the reason that it is the only definite quantity available to express this rate of agitation. It may be, to a certain extent, a measure of the rate of agitation,

*University of Illinois, Engineering Experiment Station, Bulletin No. 40.

but it is not necessarily so. The rate of agitation may be defined as the number of times per second that each particle of water comes into contact with the tube or with the film on the tube if we consider that the film is indestructible, although it is more likely that the film is in a process of continuous renewal. In the case of the present experiments, it is probable that if elaborate baffles had been placed in the tube, the heat transmission for the same velocity of flow would have been increased. Also, if the stream of water on entering the tube had been given a rotary motion the heat transmission would have been increased while the velocity through the tube remained constant."

In other words, in a feed heater the controlling resistance lies on the water side and is subject to correction by mechanical agitating or mixing devices up to the point where water ceases to flow as a uniform fluid, the point known as the critical velocity. The foregoing tests show how effective spiral retarders have proved in the Bureau type of heater.

At a water rate of 1,500 pounds per tube per hour the heat transmission was increased 16 per cent., at 2,000 pounds, 7.5 per cent., at 2,500 pounds, 14.9 per cent., and at 3,000 pounds, 14.8 per cent.

V.—Heat Transfer in Film Heaters. Schutte and Koerting Spirally-Corrugated Film Heater.

A form of heater in which the agitation is accomplished by a deformation of the tube in process of manufacture is the spirally-corrugated film heater manufactured by Schutte and Koerting. The heating element in this apparatus consists of two spirally-corrugated tubes, one within the other, the water path being the small clearance between the two. Thus the water is directed in a spiral path due to the corrugations, and for a given velocity the particles of water come more often in contact with the heating surface because they are contained within an annular space whose perimeter is large in comparison with its area.

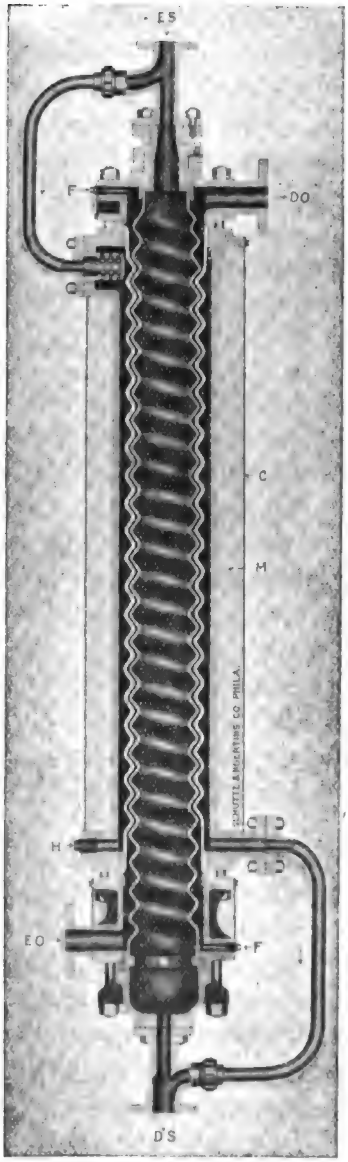


FIG. 8.

The first of these units to be tested as a feed-water heater is the same as the one whose performance as an oil heater is reported in the May, 1912, JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.* The heater, which is shown in figure 8, consists of a cylindrical composition casting containing one pair of spirally-corrugated copper tubes, one of which screws into the other. The water passes upward through the heater in a 7/64-inch nominal film between the tubes, the steam circulating downward around the outer tube and within the inner tube. The steam and water joints are external and the only failure which would not be immediately noticed would be that of a tube proper. The principal dimensions of the heater tested are given in Appendix I.

The series of tests proposed was with rates of water flow of 5,000, 10,000, 15,000 and 20,000 pounds per hour at steam pressures of 10 and 50 pounds per square inch gage. When the heater was put into service it was found that the least practicable rate of discharge was 7,500 pounds of water per hour, for at lower rates the outlet water was at such a high temperature that much of it was flashing into steam beyond the outlet control valve. Above 15,000 pounds per hour the friction drop was excessive, so the rates of discharge selected were 7,500, 10,000 and 15,000 pounds per hour.

The heat transfer was so rapid and the consequent supply of steam so great, that the steam pressures which could be used were limited by two conditions: firstly, the pressure had to be low enough that the corresponding temperature of inlet steam did not heat the water to the boiling point, and secondly, the pressure had to be high enough to keep the heater clear of water at the bottom, so as to obtain full benefit of all the heating surface and force the condensate to the weighing tanks. With atmospheric pressure at the condensed steam receiver, the inlet steam pressure varied from 11 to 31 pounds, indicating the adaptability of this apparatus with some modification to condenser service. The resistance through the

*Test of an External-Joint Film Oil Heater at the Engineering Experiment Station, Annapolis, Md., JOURNAL AM. SOC. NAV. ENG., 1912, p. 426.

heater was determined by a differential gage. For discharges below 12,000 pounds per hour the pressure drop is moderate, increasing from 7 pounds at 8,000 pounds of water per hour to nearly 40 pounds when the rate of flow is 15,000 pounds per hour.

The complete data from the tests is given in Table VI and the principal results shown graphically in figure 9. The heat transmitted in B.t.u. per square foot per hour reaches a value of 147,922 at the highest velocity. This corresponds to a heat transmission per square foot per hour per degree average temperature difference of 1,227 B.t.u. and per degree of mean temperature difference of 1,254 B.t.u., more than double that obtained at equivalent velocities in the Bureau type of heater. The true velocity of water is in this type problematical, as the spiral corrugation will produce an agitation greater than direct flow through a uniform film of the same area. In every case the axial velocity is considered, and the velocity corresponding to a reasonable friction loss would be about 250 feet per minute, or 12,930 pounds of water per hour.

Although the tests of this unit furnish no direct data to advance the theory developed in a previous section, the tests have been cited to show the high rates of heat transfer which may be attained with proper water agitation, as well as to form an introduction to more extensive tests of a similar type of heater.

The film type of heater had been selected by the Bureau of Steam Engineering for installation on the U. S. S. *Oklahoma,* conditional upon a maximum pressure drop of 10 pounds at full load. The builders thereupon constructed a small experimental heater after the design for the *Oklahoma,* containing four double-tube elements instead of sixty-six in each of two heaters for the ship, and sent the same to the Engineering Experiment Station, Annapolis, Md., where it was tested under the writer's direction in 1912. These tests covered a wide range of water velocities, steam pressures, tube arrangement and water temperatures, and the results will be presented in

TABLE VI.—DATA AND RESULTS OF TESTS ON SCHUTTE AND KOERTING FILM-OIL HEATER WHEN USED AS A FEED-WATER HEATER.

1. Test number	1	2	3	4
2. Date, May, 1912	25	25	25	27
3. Duration of test, hours	1	1	1	1
4. Barometer, ins. of mercury	29.96	29.96	29.96	30.04
5. Temperature of room, °F	84.7	88.0	88.0	79.7
6. Water through heater, lbs. per hour	7,780.5	10,897.5	11,985.5	14,807.5
7. Pressure at inlet, lbs. gage	244.8	247.6	244.7	249.2
8. Pressure at outlet, lbs. gage	238.3	234.7	230.8	215.9
9. Differential gage reading, inches of mercury	14.3	28.3	30.6	...
10. Resistance through heater, pounds per square inch	6.5	12.9	13.9	33.3
11. Temperature at inlet, °F	73.0	72.4	72.4	72.1
12. Temperature at outlet, °F	194.8	184.9	186.5	190.5
13. Temperature rise, degs. F	121.8	112.5	114.1	118.4
14. Mean specific heat	.988	.988	.988	.988
15. Heat absorbed by water, B.t.u. per hour	936,293	1,211,257	1,351,135	1,732,170
16. Steam condensed, lbs. per hour	931.0	1,229.3	1,382.5	1,830.3
17. Steam pressure at inlet, lbs. gage	11.0	16.6	19.9	27.2
18. Temperature at inlet, °F	307.5	278.7	299.1	281.2
19. Temperature corresponding to absolute pressure at inlet, degrees F	241.5	252.7	258.6	270.2
20. Superheat, degrees F	66.0	26.0	40.5	11.0
21. Temperature of condensed steam at outlet, degrees F.	211.4	206.7	212.7	222.5
22. Equivalent dry and saturated steam, lbs. per hour.	956.7	1,243.2	1,406.7	1,839.8
23. Heat supplied by steam, B.t.u. per hour	943,196	1,232,619	1,390,380	1,803,212
24. Radiation and unaccounted losses, B.t.u. per hour	7,053	21,776	39,737	72,502
25. Velocity of water through tube, feet per minute	152.8	213.7	235.1	290.6
26. Transmission in B.t.u. per square foot per hour	79,957	103,438	115,383	147,922
27. Average arithmetical temperature difference, °F	125.6	114.1	126.5	120.6
28. Transmission in B.t.u. per sq. foot of heating surface per deg. F. of average arithmetical temperature difference per hour	636.6	906.6	912.1	1,226,6
29. Mean temperature difference, degrees F	125.1	112.8	126.0	117.9
30. Heat transmission in B.t.u. per square foot of heating surface per degree F. mean temperature difference per hour	639.1	917.0	915.7	1,254.6
31. Dry steam consumed per sq. foot of heating surface per hour, pounds	81.69	106.16	120.13	157.11
32. Water heated per sq. foot of heating surface per hour, pounds	664.9	930.6	1,023.5	1,264.5
33. Water heated per pound of dry steam, pounds	8.133	8.766	8.520	8.048

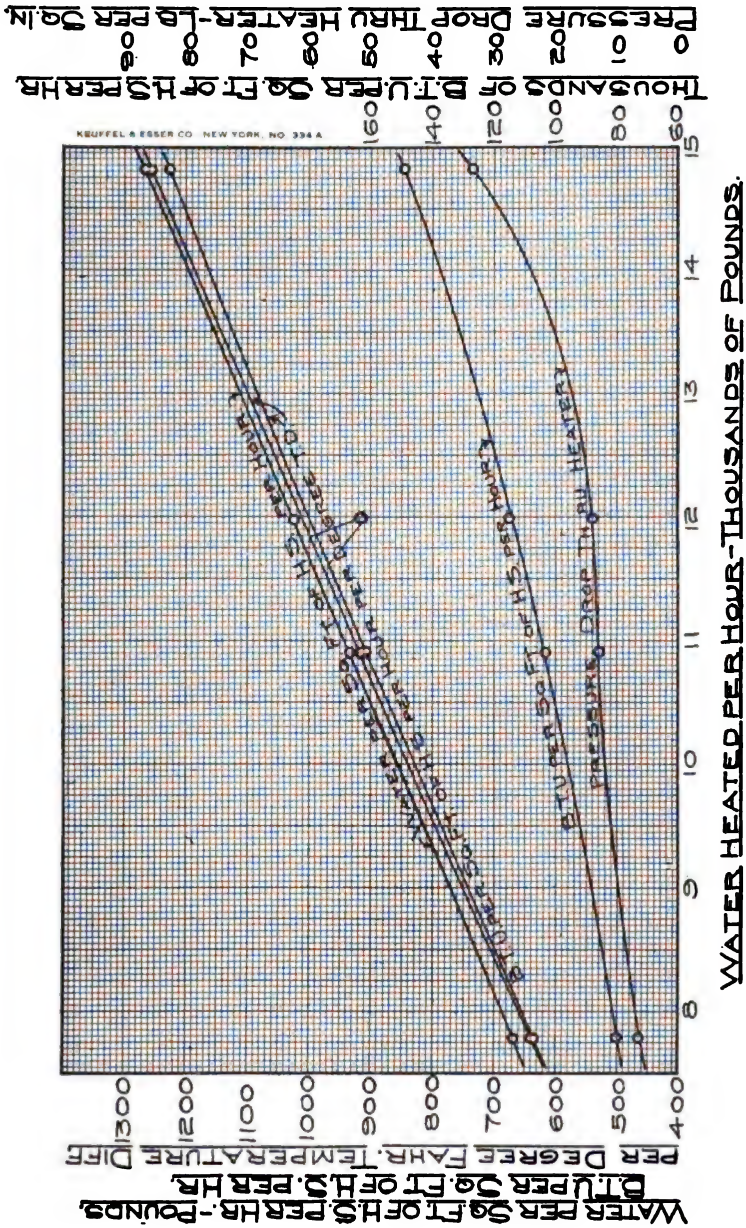
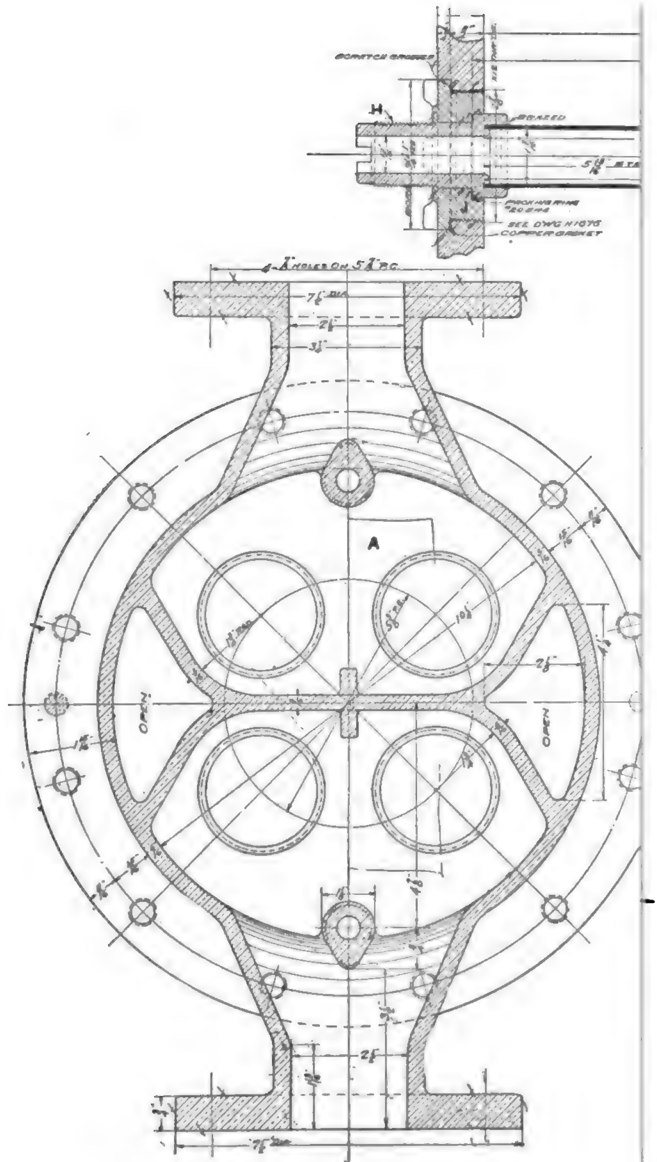

PRESSURE DROP THRU HEATER-LB. PER SQ. IN.
0 10 20 30 40 50 60 70 80 90
THOUSANDS OF B.T.U. PER SQ. FT. OF H.S. PER HR.
60 80 100 120 140 160
WATER PER SQ. FT. OF H.S. PER HOUR
B.T.U. PER SQ. FT. OF H.S. PER HOUR
PRESSURE DROP THRU HEATER
8 9 10 11 12 13 14 15
WATER HEATED PER HOUR-THOUSANDS OF POUNDS.
400 500 600 700 800 900 1000 1100 1200 1300
PER DEGREE FAHR. TEMPERATURE DIFF.
B.T.U. PER SQ. FT. OF H.S. PER HR.
WATER PER SQ. FT. OF H.S. PER HR.-POUNDS.

Fig. 9.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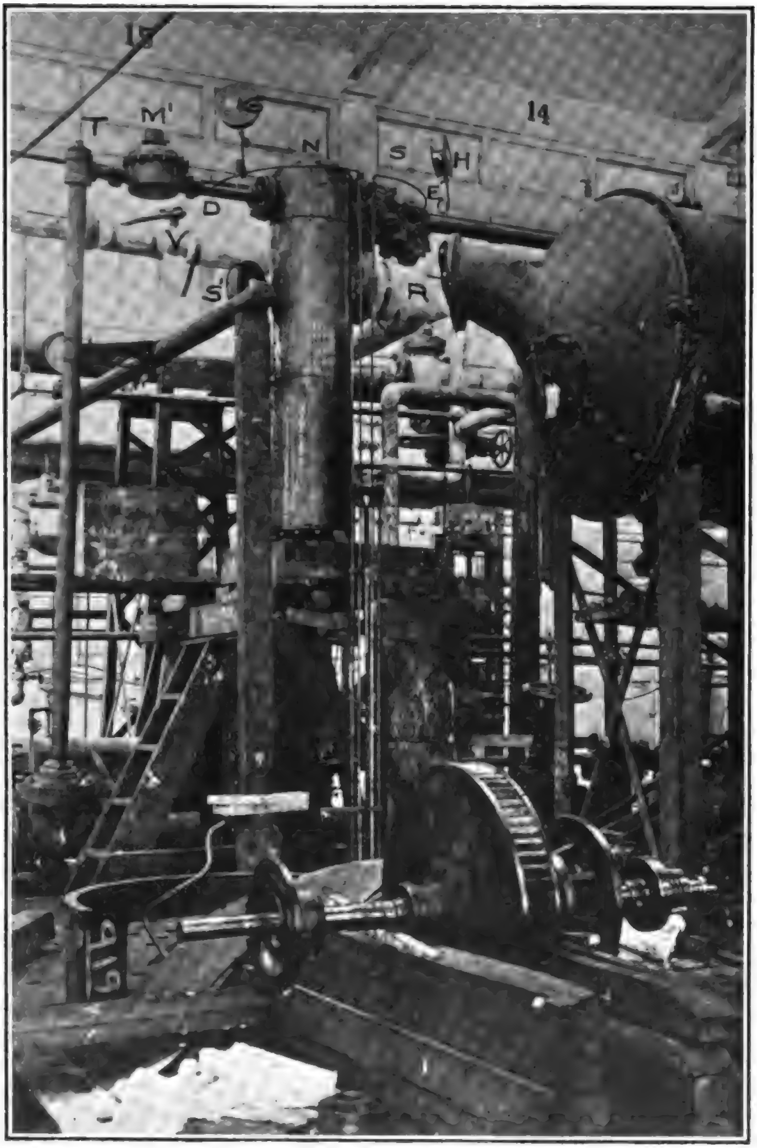


FIG. 10.

SCHUTTE AND KOERTING EXPERIMENTAL FEED-WATER HEATER ERECTED FOR TEST.

detail as containing a mass of information on this particular type of apparatus.

The experimental apparatus is shown in Plate II. There are four pairs of spirally-corrugated tubes arranged in two passes. As originally constructed the approximate water-film thickness was 7/64 inch, but there were supplied four additional inner tubes with which to replace those in the heater and obtain a film thickness of approximately 3/16 inch. The heater was apparently designed for horizontal installation, but a slight modification of the lower header permitted of vertical mounting.

The heater details give the outer tubes as No. 13 B.W.G., 3 feet 11 inches long, rigidly expanded into the headers. The inner tubes are No. 14 B.W.G., 4 feet 10¼ inches long overall and are screwed into place, the joints being made tight by locking nuts and copper packing rings. Water enters the main header through a 2½-inch opening, is directed by a center baffle down two pairs of tubes to the lower head and then upward and out. Steam connection is made through a 7-inch flange, and the condensation is removed by a 1½-inch outlet in the drain cover. Suitable lagging is provided. The principal dimensions and calculated areas of steam heating surface are given in Appendix II.

The apparatus was erected for vertical test as shown in figure 10. The water to be heated is delivered under pressure past the inlet water thermometer T, the water meter M′, and the inlet pressure gage G. Discharge pressure, maintained at about 200 pounds, is taken at gage H and discharge temperature by thermometer S. To measure the friction drop a differential mercury gage was connected across the feed line at D and E.

Steam pressure was controlled by valve V and its initial condition shown by thermometers and pressure gage attached near R. Pressures were also taken within the shell, and the temperature of condensed steam by a thermometer immersed in a drain pot at outlet. Accumulated air was withdrawn by

a pet cock in the drain pot and the water of condensation was cooled and weighed, thus obtaining a measure of the heat input as well as heat imparted to the water.

The object of the tests was to determine the effect on heat transfer of a number of variables:

(1) Velocity of feed water through the water film;
(2) Steam pressure;
(3) Temperature of inlet feed;
(4) Variation in mean film thickness;
(5) Vertical and horizontal installation.

The data and results of tests are contained in Tables VII to XI. In each of these tables the individual trials are tabulated in groups of ascending water rates at steam pressures of approximately 5, 10 and 15 pounds gage pressure. The several tables cover the following conditions:

VII. Vertical installation, 7/64-inch film thickness, inlet water temperature 66.5 degrees to 72.2 degrees F., water rates 1,750 to 10,950 pounds per tube per hour;
VIII. Horizontal installation, 7/64-inch film thickness, inlet water temperature 52.5 degrees to 55.8 degrees F., water rates 2,330 to 10,200 pounds per tube per hour;
IX. Vertical installation, 3/16-inch film thickness, inlet water temperature 70.2 degrees to 71.1 degrees F., water rates 2,640 to 10,310 pounds per tube per hour;
X. Vertical installation, 3/16-inch film thickness, inlet water temperature 87.6 degrees to 94.4 degrees F., water rates 2,660 to 12,860 pounds per tube per hour;
XI. Horizontal installation, 3/16-inch film thickness, inlet water temperature 88.6 degrees to 95.3 degrees F., water rates 2,520 to 12,220 pounds per tube per hour.

The sources of the data for the various columns of the foregoing tables and the method of obtaining the deduced data are shown in Appendix III.

					Water Data										
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	3	60	30.09	72.2	3502	70.5	216.7	146.2	1.0049	.978	584356	3.55	210.1	208.5	1.62
2	5	60	30.12	77.0	5842	72.0	207.6	136.6	1.0049	.978	780356	7.82	201.9	198.3	3.56
3	7	60	29.84	78.7	8629	71.8	198.5	126.7	1.0052	.978	1069821	14.11	213.4	207.0	6.43
4	7	60	29.73	79.3	10970	72.2	193.5	121.3	1.0052	.978	1253898	21.71	222.1	212.2	9.89
5	8	60	30.08	73.2	12586	70.2	182.8	112.6	1.0069	.973	1378900	27.36	222.9	210.4	12.46
6	9	60	30.18	69.7	14933	66.5	179.2	112.7	1.0065	.976	1598936	34.51	225.0	209.3	15.72
7	22	30	30.12	75.5	17092	67.5	180.1	112.6	1.0082	.974	1874565	——	230.5	223.7	16.8
8	22	30	30.06	82.2	21906	67.9	171.2	103.3	1.0082	.974	2204809	——	243.6	226.7	26.9
9	3	60	30.07	76.3	4287	70.9	227.8	156.9	1.0049	.978	646625	4.97	205.3	203.0	2.26
10	5	60	30.08	79.4	6368	70.8	219.0	148.2	1.0049	.978	962990	8.93	205.2	201.1	4.07
11	7	60	29.81	79.9	8955	72.2	210.5	138.3	1.0052	.978	1194390	14.40	219.7	213.1	6.56
12	8	60	30.09	71.7	10314	70.2	201.8	131.6	1.0069	.973	1320606	18.19	216.4	208.1	8.88
13	8	60	30.07	75.1	12918	70.3	192.9	122.6	1.0069	.973	1540999	28.74	226.5	213.4	13.09
14	9	60	30.16	71.1	16035	66.5	183.4	116.9	1.0065	.976	1889993	41.65	233.6	214.6	18.97
15	22	30	30.10	78.5	16583	67.6	191.3	123.7	1.0082	.974	1998002	——	227.8	212.4	15.4
16	22	30	30.04	81.2	21516	67.5	181.2	113.7	1.0082	.974	2388811	——	236.1	212.7	23.4
17	3	60	30.07	78.7	5876	71.3	240.7	169.4	1.0049	.978	642875	4.56	207.5	205.4	2.08
18	7	60	29.87	75.8	6212	72.1	229.3	157.2	1.0052	.978	954990	7.89	214.7	211.1	3.59
19	7	60	29.74	78.8	9861	72.2	219.9	147.7	1.0052	.978	1337971	16.92	219.7	212.0	7.71
20	8	60	30.08	72.2	9831	70.2	214.4	144.2	1.0069	.973	1379427	17.20	218.4	210.6	7.83
21	8	60	30.04	74.2	12950	70.4	201.5	131.1	1.0069	.973	1651860	29.60	229.4	215.9	13.48
22	9	55	30.15	72.5	15760	66.6	193.1	126.5	1.0065	.976	1945823	40.59	234.3	215.8	18.48
23	22	30	30.07	80.1	16863	67.9	199.4	131.5	1.0082	.974	2082960	——	228.8	215.0	13.8
24	22	30	30.04	78.8	21977	67.4	189.6	122.2	1.0082	.974	2546470	——	237.1	212.9	24.2

	Steam Data											
1	17	18	19	20	21	22	23	24	25	26	27	28
1	4.66	19.44	276.0	226.5	49.5	225.8	220.4	990.5	541.5	536356	12000	534.6
2	4.86	19.65	303.3	227.0	76.3	225.8	220.4	1003.4	789.0	791683	11287	828.4
3	4.78	19.44	309.4	226.5	82.9	227.2	224.2	1002.5	1071.0	1073678	4457	1114.6
4	4.76	19.57	252.2	226.2	26.0	226.0	224.4	973.4	1297.5	1263582	11704	1324.1
5	4.80	19.57	254.1	226.8	27.3	224.4	223.9	976.7	1425.0	1391798	12898	1444.1
6	4.76	19.58	298.5	226.8	71.7	226.1	224.9	996.7	1692.5	1687123	16977	1690.0
7	4.74	19.53	273.9	226.7	49.2	226.5	224.9	985.9	1964.0	1936308	61783	2011.3
8	4.75	19.52	275.9	226.6	49.3	226.6	225.2	985.7	2298.0	2265139	61130	2399.6
9	9.74	24.51	296.2	239.0	57.2	238.8	236.2	988.9	685.0	673887	24662	704.6
10	9.74	24.51	307.4	239.0	68.4	238.6	237.3	987.0	951.0	938637	15647	983.5
11	9.57	24.22	311.8	238.3	73.5	238.0	236.6	989.9	1176.0	1164122	9732	1219.1
12	9.72	24.50	309.5	239.0	70.5	237.2	235.5	989.9	1358.5	1344779	24273	1406.2
13	9.62	24.39	260.7	238.7	22.0	237.8	236.9	965.2	1628.0	1571346	30387	1643.9
14	9.80	24.61	270.0	239.2	30.8	236.7	236.2	970.4	1931.5	1874388	44375	1961.2
15	9.90	24.68	273.2	239.4	33.8	238.4	237.0	971.0	2120.0	2058920	60318	2155.8
16	9.90	24.66	281.2	239.3	41.9	240.2	238.7	973.0	2507.0	2439311	56500	2559.4
17	14.66	29.43	300.0	249.2	50.8	248.7	246.8	972.8	672.1	653819	11644	683.4
18	14.46	29.14	311.9	248.7	63.2	249.3	246.8	978.5	991.5	970183	15193	1023.2
19	14.48	29.09	260.6	248.6	12.0	249.2	247.5	953.1	1421.0	1354355	26784	1489.6
20	14.46	29.23	306.0	248.8	57.2	247.9	246.2	976.2	1434.0	1399871	20454	1475.4
21	14.40	29.16	267.4	248.7	18.7	246.4	246.2	957.7	1762.0	1687467	35607	1778.7
22	14.45	29.26	276.9	248.9	28.0	247.5	246.9	961.6	2077.6	1997820	52997	2107.1
23	14.60	29.37	278.9	249.1	29.8	248.6	247.3	962.0	2285.0	2140450	57490	2258.6
24	14.60	29.36	283.8	249.1	34.7	248.7	247.8	963.9	2683.0	2586144	39374	2730.2

	Deduced Data									
1	29	30	31	32	33	34	35	36	37	38
1	90.3	24582	82.9	52.8	296.5	465.4	158.4	164.2	97.8	1751
2	190.3	36585	87.7	65.5	417.2	558.6	140.1	273.9	98.6	2921
3	222.0	50185	91.3	74.2	540.0	675.5	125.2	404.5	99.6	4315
4	271.9	59782	93.3	78.3	630.0	750.8	124.3	495.5	99.1	5285
5	323.2	64643	100.3	88.7	644.5	788.7	114.7	590.0	99.1	6255
6	373.1	74559	103.9	92.9	721.3	806.6	116.3	681.3	98.2	7267
7	438.2	87079	102.9	91.7	854.0	958.3	117.7	801.3	96.8	8546
8	561.6	103324	107.0	98.2	965.6	1052.2	107.4	1087.0	97.3	10993
9	108.7	30408	89.6	57.9	339.4	525.1	166.7	198.2	96.5	2114
10	163.8	43870	94.1	69.6	459.8	621.6	154.4	298.5	98.3	3184
11	219.6	54118	96.9	77.4	558.5	699.2	142.8	400.1	99.2	4268
12	264.8	63910	103.0	87.0	601.1	711.6	136.3	483.5	98.2	5297
13	331.7	72240	107.1	94.2	674.5	766.9	127.4	605.6	98.1	6459
14	411.8	85788	114.2	103.5	751.2	848.9	122.3	751.9	97.6	8080
15	485.1	95666	109.9	97.0	852.3	965.6	130.0	777.4	97.1	8292
16	551.6	111706	114.9	104.9	972.2	1064.9	119.0	1008.7	97.9	10758
17	99.7	30105	93.2	55.7	383.0	540.4	177.9	181.7	98.2	1938
18	159.8	44770	98.0	71.2	456.8	638.8	164.7	291.2	98.4	3206
19	238.3	68705	102.5	81.3	611.8	771.2	154.4	434.2	98.8	4631
20	252.4	64667	106.5	87.5	607.2	739.1	150.1	460.9	98.5	4916
21	338.5	77459	112.7	98.7	687.1	785.0	137.4	607.1	97.9	6475
22	404.7	91220	119.0	106.8	766.5	854.1	133.7	738.8	97.4	7880
23	436.9	97649	135.4	101.7	846.2	960.2	138.9	762.4	97.3	8132
24	548.5	119393	180.6	109.5	990.0	1090.3	127.6	1003.1	98.5	10699

TABLE VII.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER.—TABLE OF DATA AND RESULTS. 7/64-INCH WATER FILM.—VERTICAL ARRANGEMENT.

					Water Data.											Steam Data.												Reduced Data.									
Number of Test.	Date, November, 1912.	Duration of Test, Minutes.	Barometer, Inches of Mercury.	Room Temperature, Degrees F.	River Water Pumped through Heater, Pounds per Hour.	Temperature at Inlet, Degrees F.	Temperature at Outlet, Degrees F.	Temperature Rise, Degrees F.	Density of River Water.	Specific Heat of River Water.	B.T.U. Absorbed by Water per Hour.	Friction Drop, Inches of Mercury.	Water Pressure at Inlet, Pounds per Sq.In.	Water Pressure at Outlet, Pounds per Sq.In.	Friction Drop, Pounds per Sq.In.	Pressure at Inlet, Pounds Gage.	Pressure at Inlet, Pounds Absolute.	Temperature at Inlet, Degrees F.	Temperature corresponding to Pressure.	Degrees Superheat.	Temperature at Outlet of Heater.	Temperature of Condensed Steam, Degrees F.	Heat Rejected per Pound of Steam, B.T.U.	Steam Condensed per Hour, Pounds.	B.T.U. Rejected by Steam per Hour.	Radiation, B.T.U. per Hour.	Equivalent Dry and Saturated Steam, Pounds per Hour.	Axial Velocity, Feet per Minute.	B.T.U. Transmitted per Sq.Ft. per Hour.	Average Arithmetical Temperature Difference.	Mean Temperature Difference.	B.T.U. per Sq.Ft. per Hour per Degree Arithmetical Mean Temp. Difference.	B.T.U. per Sq.Ft. per Hour per Degree Mean Temp. Difference.	Pounds of Dry Steam per 1000 Pounds of Water Heated.	Water Heated per Sq.Ft. of Heating Surface per Hour.	Efficiency as a Heater.	Pounds of Water per Tube per Hour.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
37	18	30	30.14	64.1	4727	55.4	198.2	142.8	1.008	.970	633580	3.15	225.4	223.1	2.34	4.73	19.53	301.5	226.7	74.8	224.1	221.6	1001.2	682	682018	48438	706.6	120.8	30630	99.9	79.6	306.6	384.8	143.8	221.1	95.7	2587
38	18	30	30.12	70.0	9672	54.7	180.1	125.4	1.008	.970	1176478	18.67	201.9	193.4	8.49	4.70	19.49	305.5	226.6	78.9	225.6	223.8	998.9	1147	1145738	-30740	1191.7	247.8	55293	109.2	93.9	505.1	575.1	123.2	453.4	—	4836
39	18	30	30.14	72.0	14993	54.7	167.6	112.9	1.008	.970	1643917	—	237.5	219.4	18.1	5.45	20.25	265.5	228.7	36.8	229.3	228.8	976.7	1702	1662344	20427	1732.6	384.1	76074	117.5	107.9	655.1	723.4	115.6	702.9	98.8	7496
40	19	30	30.06	70.0	20402	54.6	153.0	98.4	1.008	.970	1947326	—	237.9	206.9	31.0	5.45	20.21	269.9	228.5	41.4	228.1	227.8	979.9	2025	1984298	36972	2065.5	522.6	91291	124.7	117.9	732.1	774.3	101.2	956.4	98.1	10001
41	18	30	30.14	65.9	4729	55.2	213.8	158.6	1.008	.970	727514	3.49	210.3	208.0	2.32	10.12	24.92	310.4	239.9	70.5	239.2	237.7	988.0	751	741988	14474	777.3	126.1	34106	105.4	81.0	323.6	421.1	164.4	221.7	98.0	2565
42	18	30	30.13	70.9	9572	54.7	193.1	138.4	1.008	.970	1285016	17.97	215.9	207.7	8.18	10.30	25.09	289.3	240.7	48.6	242.0	240.7	974.8	1432	1395914	110898	1466.4	245.2	60042	116.8	101.5	515.8	593.5	133.2	448.7	—	4706
43	19	30	30.09	65.1	14417	55.8	174.3	118.5	1.008	.970	1658513	37.22	223.1	206.2	16.94	10.47	25.25	296.0	240.6	55.4	240.3	238.4	980.3	1750	1715986	57473	1799.0	369.6	77460	125.5	115.6	617.2	670.1	124.7	696.3	96.7	7223
44	19	30	30.06	69.0	19600	53.5	159.6	106.1	1.008	.970	2017173	—	237.0	205.2	31.8	9.83	24.59	277.5	239.2	38.3	239.4	238.9	971.1	2136	2074270	57097	2278.7	502.0	94365	130.6	125.2	723.2	755.3	111.2	918.8	97.2	9800
45	18	30	30.14	68.0	4792	55.5	222.2	166.7	1.008	.970	774855	3.80	205.2	202.6	2.64	14.75	29.55	311.4	249.5	61.9	249.1	247.4	977.0	786	767922	-6933	810.4	122.8	36385	110.6	85.0	328.4	427.4	169.1	224.6	—	2596
46	18	30	30.14	71.3	10339	54.8	198.8	144.0	1.008	.970	1440147	20.56	210.6	201.2	9.35	15.16	29.96	275.2	250.2	25.0	250.1	249.3	957.7	1542	1476774	36627	1560.5	264.8	67701	123.4	107.8	548.6	628.0	150.9	454.7	97.8	5270
47	19	30	30.09	68.0	14452	54.6	183.4	128.8	1.008	.970	1805570	36.42	221.0	204.4	16.57	15.41	30.19	271.6	250.7	20.9	250.3	250.2	955.4	1934	1847744	42174	1955.3	370.2	84645	131.7	120.4	642.7	703.0	135.3	677.5	97.7	7206
48	19	30	30.01	71.7	19459	52.5	165.4	112.9	1.008	.970	2235391	—	234.9	204.9	30.0	15.00	29.74	279.7	249.8	29.9	250.0	249.7	959.9	2384	2292412	57021	2328.9	489.4	100267	140.8	132.9	711.0	755.3	118.9	914.1	97.4	9750

TABLE VIII.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. TABLE OF DATA AND RESULTS. 7/64-INCH WATER FILM.—HORIZONTAL ARRANGEMENT.

Number of Test.	Date — Oct., 1912.	Duration of Test.	Barometer — Inches of Mercury.	Room Temperature, Degrees F.
1	2	3	4	5
25	10	60	29.99	79.0
26	11	60	30.05	81.3
27	11	60	30.02	82.5
28	14	60	30.16	67.3
29	11	60	30.07	76.3
30	11	60	30.03	82.5
31	12	60	29.97	73.2
32	14	60	30.15	67.1
33	11	60	30.06	77.9
34	11	60	30.02	82.7
35	12	60	29.89	79.4
36	14	60	30.24	66.7

Water Data.

Number of Test.	River Water Pumped Through Heater, Pounds per Hour.	Temperature at Inlet, Degrees F.	Temperature at Outlet, Degrees F.	Temperature Rise, Degrees F.	Density of River Water.	Specific Heat of River Water.	B.T.U. Absorbed by Water per Hour.	Friction Drop, Inches of Mercury.	Water Pressure at Inlet, Pounds per Sq.In.	Water Pressure at Outlet, Pounds per Sq.In.	Friction Drop, Pounds per Sq.In.
1	6	7	8	9	10	11	12	13	14	15	16
25	5332	70.1	207.9	137.8	1.0074	.9780	722810	2.78	213.7	212.4	1.27
26	10659	70.2	193.0	122.8	1.0074	.9780	1272869	9.12	229.6	225.4	4.25
27	15373	70.7	178.7	108.0	1.0074	.9780	1613802	17.70	227.5	219.4	8.06
28	20627	71.1	166.8	95.7	1.0077	.9714	1917550	29.64	227.7	214.2	13.90
29	5279	71.1	220.1	149.0	1.0074	.9780	764349	2.70	226.9	225.7	1.23
30	10740	70.3	202.6	132.3	1.0074	.9780	1370680	9.34	229.7	215.4	4.25
31	15552	70.2	188.3	118.1	1.0077	.9714	1762123	17.50	223.0	215.0	7.97
32	20300	71.1	176.1	105.0	1.0077	.9714	2070537	29.16	229.4	216.1	13.28
33	5348	71.1	230.4	159.3	1.0074	.9780	837149	2.76	228.7	227.4	1.26
34	10615	70.6	210.7	140.1	1.0074	.9780	1445324	9.02	227.6	223.5	4.11
35	14900	70.6	198.0	127.4	1.0077	.9714	1843975	16.40	223.1	215.6	7.47
36	20614	71.1	182.2	111.1	1.0077	.9714	2224711	29.20	226.8	213.5	13.30

Steam Data.

Number of Test.	Pressure at Inlet, Pounds Gage.	Pressure at Inlet, Pounds Absolute.	Temperature at Inlet, Degrees F.	Temperature Corresponding to Pressure.	Degrees Superheat.	Temperature at Outlet of Heater, Degrees F.	Temperature of Condensed Steam, Degrees F.	Heat Rejected per Pound Steam, B.T.U.	Steam Condensed per Hour, Pounds.	B.T.U. Rejected by Steam per Hour.	Radiation, B.T.U. per Hour.	Equivalent Dry and Saturated Steam, Pounds per Hour.
1	17	18	19	20	21	22	23	24	25	26	27	28
25	4.58	19.32	291.6	226.1	65.5	222.9	220.2	998.2	745.0	743659	21449	768.8
26	4.70	19.46	321.0	226.5	84.5	226.2	226.3	1001.3	1295.5	1297184	24315	1309.4
27	4.63	19.39	261.1	226.3	34.8	225.2	225.4	978.6	1701.5	1665088	51287	1730.7
28	4.68	19.54	292.2	226.7	65.5	226.5	225.1	993.9	1968.0	1955208	37658	2052.3
29	10.18	24.95	305.1	240.0	65.1	236.5	236.1	987.2	805.0	794696	30247	851.0
30	9.90	24.65	260.8	239.3	21.5	238.0	238.0	964.5	1457.0	1405277	34597	1473.4
31	9.87	24.60	293.9	239.2	54.7	236.4	236.2	981.8	1850.5	1816882	55608	1900.5
32	9.82	24.67	277.1	239.4	37.7	238.8	237.3	972.6	2294.0	2233084	65547	2353.2
33	14.58	29.34	310.8	249.1	61.7	247.9	247.9	976.8	865.5	845420	10271	892.6
34	14.53	29.28	263.5	248.9	14.6	248.2	248.2	959.9	1557.5	1485699	40375	1569.0
35	14.52	29.21	297.5	248.8	48.7	247.5	247.6	970.8	1947.5	1890633	46658	1995.8
36	14.50	29.35	283.1	249.1	36.0	248.9	247.1	963.3	2371.5	2283809	64498	2425.1

Reduced Data.

Number of Test.	Axial Velocity, Feet per Minute.	B.T.U. Transmitted per Sq.Ft. per Hour.	Average Arithmetical Temperature Difference.	Mean Temperature Difference.	B.T.U. per Sq.Ft. per Hour per Degree Arith. Temp. Dif.	B.T.U. per Sq.Ft. per Hour per Degree Mean Temp. Dif.	Pounds of Dry Steam per 1000 Pounds of Water Heated.	Water Heated per Sq.Ft. of Heating Surface per Hour.	Efficiency of Heater.	Pounds of Water per Tube per Hour.
1	29	30	31	32	33	34	35	36	37	38
25	90.0	57526	87.1	64.1	405.4	552.0	142.5	369.7	97.1	2696
26	177.9	60214	94.9	73.7	635.6	780.6	126.6	582.2	98.1	5330
27	256.2	70915	101.6	91.2	776.7	863.9	112.5	752.7	96.9	7687
28	344.3	93768	107.7	100.2	870.6	938.8	98.4	1008.7	98.1	10313
29	88.1	57365	94.4	69.7	536.0	536.4	157.4	258.1	96.2	2640
30	179.3	67086	103.3	87.5	648.8	766.0	137.2	525.2	97.5	5370
31	256.2	86123	109.9	98.4	783.6	875.2	123.8	750.7	96.9	7676
32	338.8	101249	115.8	107.4	874.3	942.7	120.1	992.7	97.0	10150
33	89.2	40448	98.9	70.7	411.5	572.1	167.1	261.2	97.8	2671
34	177.2	70686	108.2	90.9	653.3	777.6	147.8	519.1	97.3	5308
35	248.6	90270	114.5	101.5	787.5	888.4	133.9	728.6	97.5	7450
36	344.0	108788	122.4	113.5	888.8	958.5	117.1	1008.0	97.2	10307

TABLE IX.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. 3/16-INCH WATER FILM.—VERTICAL ARRANGEMENT.

Number of Test.	Date — Dec. 1912.	Duration of Test, Minutes.	Barometer, Inches of Mercury.	Room Temperature, Degrees F.	Water Data. Water Heated, Pounds per Hour.	Temperature at Inlet, Degrees F.	Temperature at Outlet, Degrees F.	Temperature Rise, Degrees F.	Density of Water.	Specific Heat of Water.	B.T.U. Absorbed by Water per Hour.	Friction Drop, Inches of Mercury.	Water Pressure at Inlet, Pounds per Sq. In.	Water Pressure at Outlet, Pounds per Sq. In.	Friction Drop, Pounds per Sq. In.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
65	26	30	30.27	61	5862	88.9	207.5	118.6	1.0079	.974	677061	3.21	204.8	203.3	1.46
66	26	30	30.19	68	9964	90.5	198.8	108.3	1.0079	.974	1051202	7.30	211.2	207.9	3.32
67	26	30	30.17	69	14885	89.5	187.1	97.6	1.0079	.974	1409858	14.59	213.5	206.9	6.64
68	27	30	29.47	64	19137	90.3	178.7	88.4	1.0079	.974	1647696	22.60	214.3	204.0	10.29
69	27	30	29.45	65	24750	94.0	171.3	77.3	1.0079	.974	1863675	36.02	218.0	201.6	16.40
70	26	30	30.23	66	5680	90.4	218.7	128.3	1.0079	.974	712000	2.90	205.3	204.0	[illegible]
71	26	30	30.19	68	10904	88.1	206.2	118.1	1.0079	.974	1184960	7.71	212.3	208.8	3.51
72	27	30	29.51	63	15858	90.8	194.5	103.7	1.0079	.974	1541058	15.19	210.8	203.9	6.92
73	27	30	29.43	65	19387	87.6	184.8	97.2	1.0079	.974	1835949	23.10	214.7	204.2	10.52
74	27	30	29.47	65	24857	90.7	178.3	87.6	1.0079	.974	2118996	36.54	224.4	207.8	16.64
75	26	30	30.21	66	5313	94.0	230.8	136.8	1.0079	.974	707692	2.98	203.9	202.7	1.17
76	26	30	30.18	68	10060	88.1	214.0	125.9	1.0079	.974	1257876	7.65	214.1	210.6	3.48
77	27	30	29.47	64	15450	98.2	201.9	109.7	1.0079	.974	1650060	15.19	210.5	203.6	6.92
78	27	30	29.44	65	19694	94.4	195.9	101.5	1.0079	.974	1948737	23.72	214.1	203.3	10.80
79	27	30	29.49	65	23716	91.8	183.9	92.1	1.0079	.974	2306725	38.93	223.8	206.2	17.55

Number of Test.	Steam Data. Pressure at Inlet, Pounds Gage.	Pressure at Inlet, Pounds Absolute.	Temperature at Inlet, Degrees F.	Temperature Corresponding to Pressure, Degrees F.	Degrees Superheat.	Temperature at Outlet of Heater, Degrees F.	Temperature of Condensed Steam, Degrees F.	Heat Delivered per Pound of Steam, B.T.U.	Steam Condensed per Hour, Pounds.	B.T.U. Delivered by Steam per Hour.	Radiation, B.T.U. per Hour.	Equivalent Dry and Saturated Steam, Pounds per Hour.
1	17	18	19	20	21	22	23	24	25	26	27	28
65	4.91	19.77	309.1	227.3	81.8	227.2	225.7	1000.8	707	707566	30505	735
66	4.93	19.76	311.5	227.3	84.2	227.9	227.6	1000.0	1074	1074000	22798	1118
67	4.76	19.57	239.8	226.9	12.9	228.2	227.5	966.2	1496	1445435	35577	1505
68	4.80	19.27	267.7	225.9	41.8	226.5	226.6	980.5	1721	1687441	39745	1737
69	4.83	19.29	279.6	226.0	53.6	226.1	225.8	986.9	1945	1919521	55846	1996
70	9.89	24.74	311.2	239.5	71.7	239.0	237.9	988.2	759	750044	40044	786
71	9.83	24.66	296.6	239.3	57.3	239.8	238.9	980.2	1239	1214468	29508	1275
72	9.76	24.25	307.9	238.4	69.5	237.9	237.4	987.2	1599	1578933	37475	1635
73	9.73	24.18	263.6	238.2	25.4	237.6	237.5	966.0	1947	1880802	44833	1972
74	9.72	24.19	268.1	238.2	29.9	238.0	237.7	968.1	2247	2175321	56725	2281
75	14.93	29.76	313.6	249.8	63.8	250.4	249.0	976.9	760	742444	34752	784
76	15.11	29.93	254.3	250.2	4.1	250.6	249.4	948.1	1369	1297949	40073	1372
77	15.01	29.48	272.4	249.3	23.1	249.5	248.5	957.7	1765	1690341	40281	1786
78	15.07	29.49	305.4	249.3	56.1	249.4	247.9	974.2	2057	2003989	56192	2127
79	15.00	29.48	283.8	249.3	34.5	249.9	248.9	962.8	2463	2371376	64651	2507

Number of Test.	Deduced Data. Actual Velocity, Feet per Minute.	B.T.U. Transmitted per Sq.Ft. per Hour.	Average Arithmetical Temperature Difference.	Mean Temperature Difference.	B.T.U. per Sq.Ft. per Hour per Degree Average Temperature Difference.	B.T.U. per Sq.Ft. per Hour per Degree Mean Temperature Difference.	Pounds of Dry Steam per 1000 Pounds of Water Heated.	Water Heated per Sq.Ft. of H.S. per Hour.	Efficiency as a Heater.	Pounds of Water per Sq.Ft. per Hour.
1	29	30	31	32	33	34	35	36	37	38
65	98.1	33108	79.1	61.0	418.6	542.8	125.4	286.7	95.7	2931
66	166.7	51404	82.6	69.2	622.3	742.8	112.2	487.2	97.9	4982
67	248.1	68942	88.6	78.8	778.1	874.9	101.5	724.9	97.5	7423
68	380.3	80573	91.4	83.8	881.5	961.5	91.8	935.8	97.6	9569
69	424.5	91134	93.3	87.8	976.8	1038.0	80.6	1210.3	97.1	12375
70	95.1	34729	84.9	65.1	408.9	553.3	138.4	277.8	94.7	2840
71	172.4	57345	92.1	77.7	629.2	745.8	123.7	503.9	97.6	5152
72	253.4	75358	95.7	85.5	787.4	881.4	108.5	746.1	97.6	7689
73	384.3	89778	102.0	93.7	880.2	958.1	101.7	948.0	97.6	9694
74	435.7	103600	103.7	97.2	999.0	1065.8	91.8	1214.5	97.4	12439
75	89.0	34666	87.4	65.0	385.9	538.4	147.6	259.8	95.3	2657
76	171.7	61510	99.1	84.0	640.7	732.1	133.7	501.7	96.9	5190
77	258.6	80688	102.2	91.6	789.5	880.9	115.6	755.5	97.6	7725
78	389.8	95244	104.1	93.9	934.9	999.4	107.5	963.0	97.2	9847
79	430.5	112799	111.4	104.8	1022.6	1076.3	97.5	1257.5	97.3	12858

TABLE X.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. TABLE OF DATA AND RESULTS.—3/16-INCH WATER FILM.—VERTICAL ARRANGEMENT.

					Water Data.											Steam Data.												Derived Data.									
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38
50	17	30	29.92	69	5067	91.6	205.6	114.0	1.0058	0.980	580324	2.67	204.4	203.2	1.22	4.86	19.56	307.2	226.8	80.4	227.0	226.2	999.4	618	617689	29305	642.4	88.3	28769	78.2	61.6	367.9	467.0	122.0	257.6	95.3	2694
51	17	30	29.90	69	10557	92.2	192.4	100.2	1.0058	0.980	1036697	8.04	208.7	205.0	3.66	4.76	19.46	312.6	226.9	86.1	225.7	225.4	1002.9	1030	1033045	16348	1094.5	177.1	50694	84.2	73.1	602.1	693.5	103.7	516.2	98.4	5279
52	18	30	29.90	68	15370	91.4	183.6	92.2	1.0058	0.980	1389448	16.08	212.3	205.0	7.32	4.67	19.56	284.4	226.2	58.2	225.9	225.9	989.0	1432	1415259	25811	1472.0	257.8	67944	88.7	80.1	766.0	848.2	93.8	751.6	98.2	7685
53	18	30	29.86	70	20216	90.2	173.0	82.8	1.0058	0.980	1639538	25.46	215.3	203.7	11.59	4.70	19.37	268.8	226.2	42.6	224.7	224.8	982.8	1676	1647173	7635	1711.2	339.0	80272	94.6	88.1	847.5	910.0	84.6	988.6	99.5	10108
54	18	30	29.78	72	24248	90.8	167.7	76.9	1.0058	0.980	1820899	35.35	219.1	203.0	16.09	4.71	19.34	284.2	226.2	58.0	223.7	224.1	990.7	1880	1862536	34277	1933.6	406.7	89404	96.9	91.4	922.6	978.2	79.7	1185.7	98.2	12124
55	17	30	29.92	69	5270	95.3	217.9	122.6	1.0058	0.980	620917	2.75	205.5	204.2	1.29	9.84	24.54	311.3	239.0	72.3	239.8	239.3	986.9	690	641485	20568	673.6	86.8	30363	82.4	63.9	368.5	475.2	130.3	252.8	96.8	2585
56	17	30	29.92	69	10050	92.9	206.0	113.1	1.0058	0.980	1123540	7.37	207.7	204.3	3.36	9.78	24.48	314.0	238.9	75.1	239.6	239.4	988.1	1150	1136325	12775	1193.2	168.6	54452	89.4	75.0	609.1	726.0	118.7	491.4	98.0	5085
57	18	30	29.88	70	15274	92.3	192.5	100.2	1.0058	0.980	1499907	15.43	220.1	213.1	7.03	9.79	24.47	258.6	238.9	19.7	238.4	238.5	962.6	1572	1513207	13300	1587.4	256.2	73345	96.5	87.1	760.1	842.1	103.9	746.9	99.1	7637
58	18	30	29.86	70	20306	91.1	183.6	92.5	1.0058	0.980	1839894	25.85	215.0	203.2	11.77	9.87	24.54	263.8	239.0	26.8	238.6	239.1	965.4	1960	1892184	52290	1986.5	344.0	90349	101.6	94.2	895.2	965.5	96.9	1002.7	98.3	10253
59	18	30	29.77	73	24435	88.6	175.7	87.1	1.0058	0.980	2086740	35.73	225.1	208.8	16.27	9.78	24.41	272.1	237.8	34.3	238.1	238.9	968.8	2182	2113928	27173	2219.5	409.7	102040	103.6	99.5	966.3	1065.5	90.8	1194.8	98.7	12218
60	17	30	29.58	69	5048	92.3	228.5	136.2	1.0058	0.980	673908	2.73	209.6	208.4	1.24	14.89	29.59	314.0	249.5	64.5	249.4	250.3	975.9	720	702648	28740	743.8	84.7	32934	89.1	67.6	369.9	487.5	147.3	246.8	95.9	2524
61	18	30	29.91	68	10100	92.8	214.0	121.2	1.0058	0.980	1200890	7.43	205.5	202.1	3.38	14.97	29.67	315.0	249.7	65.5	249.2	250.0	976.7	1256	1226735	25845	1298.0	169.5	58724	96.3	81.9	609.8	717.0	128.5	493.9	97.9	5050
62	18	30	29.87	70	15689	92.2	202.4	110.2	1.0058	0.980	1694412	15.92	215.9	208.6	7.25	14.98	29.65	260.3	249.6	10.7	250.0	251.1	949.2	1808	1716154	21742	1818.0	268.4	83836	103.3	92.5	808.1	895.7	115.9	767.2	98.7	7845
63	18	30	29.78	72	20431	92.5	190.4	98.9	1.0058	0.980	1979764	25.83	218.9	207.1	11.76	14.88	29.51	279.0	249.4	29.6	248.9	250.0	959.4	2096	2010902	31138	2127.6	342.7	96810	108.4	100.3	893.1	965.2	104.1	999.1	98.5	10216
64	18	30	29.76	73	24126	92.6	184.4	91.8	1.0058	0.980	2271930	36.64	227.4	211.6	15.78	15.04	29.66	279.2	249.7	29.5	248.9	248.8	960.7	2300	2309610	37870	2334.7	406.7	106379	111.2	104.4	954.9	1027.0	96.8	1279.8	98.9	12063

TABLE XI.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. TABLE OF DATA AND RESULTS.
3/16-INCH WATER FILM.—HORIZONTAL ARRANGEMENT.

Attempts were made to obtain the temperature of the water at the reversal of flow in the lower head. No practical way was found to get an accurate measure of this temperature, hence a temperature gradient could not be constructed, and the value of n, the exponent in the heat transfer relation, had to be determined indirectly. Inasmuch as the temperature rise on test was usually in excess of 100 degrees, the average temperature difference would not be close enough to the mean temperature to obtain the value of the exponent directly from a logarithmic plot. The mean temperature difference for all tests was computed for assumed values of n from 0.7 to 1.0 and the resulting values of K compared. It was found that for a value of $n = 0.9$, the values of K for one water velocity and varying steam and inlet-water temperatures were substantially constant. Hence for $n = 0.9$ the relation

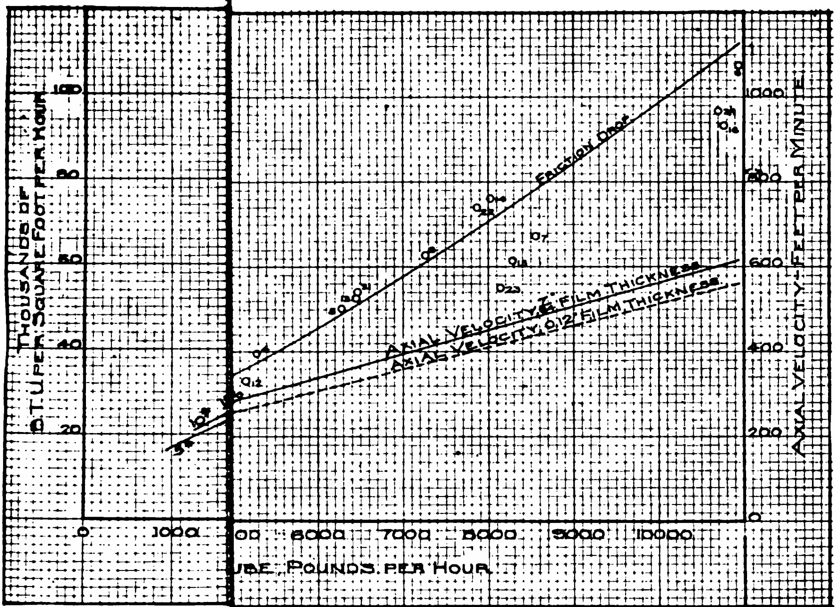
$$Q = KS(t_s - t)^n$$

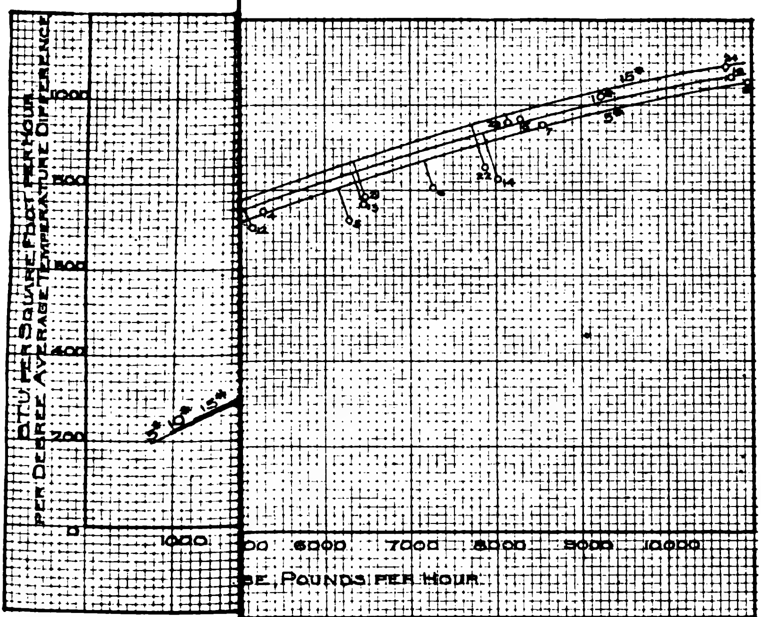
was satisfied, and for the film-heater heat transfer is proportional to the 0.9 power of the temperature difference. The values of K for 3/16-inch film and various water rates were determined as:

Water per tube, pounds per hour.	Coefficient, K.
2,500	706
5,000	1,049
7,500	1,460
10,000	1,493

The above illustrates clearly that slight advantage can be gained from water-film agitation above 7,500 pounds per tube per hour. This must be considered as the critical velocity for such construction, and higher rates would produce no increased heat transfer but would result in excessive friction losses.

The general test results were originally based on logarithmic mean temperature difference, and these are cited here as





CURVES FOR TEST GINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

showing the characteristics in the same degree as the exponential temperature difference. These results are shown graphically in figures 11 and 12 for the 7/64-inch film, and in figures 13 and 14 for 3/16-inch film. It is to be noted that from figures 11 and 13 the rates of heat transmission are nearly the same with 7/64-inch as with 3/16-inch films at the same rate of flow per tube. This suggested the plan of plotting the results as a function of the pounds of water per square foot per hour, since the heating surface for the 3/16-inch film is slightly less than for the 7/64-inch film. The resulting curves are presented in figure 15 for a steam pressure of 10 pounds gage. It appears that for the same rate of surface flow—pounds of water per square foot of heating surface per hour—the 7/64-inch film shows from 3½ to 4¼ per cent. greater heat transmission for equal rates of flow than the 3/16-inch film. It accomplishes this increase, however, with a friction loss two and one-half times as great.

Considering now the relation between vertical and horizontal heaters of the same film thickness: At 500 pounds of water per unit of surface the horizontal heater shows rates of heat transmission of 720 B.t.u. per hour per degree temperature difference as against 910 B.t.u. for the vertical heater, an advantage of 26.4 per cent. in favor of vertical installation. Care was taken to give the horizontal heater sufficient pitch to cause the water of condensation to flow freely from the tubes. This did not entirely accomplish the desired result, as the weights of condensed steam for the several intervals of any one test varied between wide limits, whereas in the vertical heater these quantities remained remarkably constant. However, the great difference between the performance of the horizontal and vertical heater cannot be entirely accounted for by the accumulation of water of condensation in the lower part of the corrugations of the inner tube, such as would render partly inactive less than 5 per cent. of the total surface. A reason is, however, to be found in the accumulation of air in the steam space. Air at a given pressure has a greater density than steam at the same pressure and corresponding tempera-

ture of saturation. Hence the air will accumulate over collected water, and the usual precautions of blowing through pet cocks in the covers would be ineffective with such an installation. The presence of this air would actively reduce the transmitting surface, and this, taken with the irregular flow of water of condensation, would explain the variation. In a subsequent test a more efficient disposal of air, attendant, however, with considerable loss of steam, resulted in rates of heat transmission of about 5 per cent. less for horizontal than for vertical installation.

There is a noticeable lack of uniformity between the friction losses for high and low rates of flow with 7/64-inch-film vertical heater, and between the results with vertical and horizontal heater, 7/64-inch film. With a small film the clearances cannot be considered to remain fixed for any period of time. Friction is altered by simply turning steam on a cold heater, hence the clearances and adjustments of the tubes are altered materially by the action of heat as well as by possible interchanges of units when the apparatus is assembled after cleaning.

The 3/16-inch film, on the other hand, gives uniform values of friction loss under widely varying temperature limits, and possesses the further great advantage of replacement of tubes without danger of change in adjustment sufficient to modify the nominal film thickness. The greater uniformity in heat transfers with this film over the 7/64-inch film is likely due to the same cause. Uniform film means uniform agitation and ability to reproduce results, whereas variations in clearance produce temporary obstructions which increase agitation and hence modify heat transfer.

The standardization of the 3/16-inch-film vertical heater is therefore advisable from the important considerations of minimum friction loss, equal heat transfer per unit of heating surface, efficient removal of water of condensation and non-condensible gases from contact with the heating surface and the ability to maintain uniform film thickness when assembling the units.

VI.—Design Curves for Feed Heaters.

The primary object of such tests as have been cited is to obtain data of value in design. Following the suggestions of the Bureau of Steam Engineering these data have been incorporated in a number of design curves which permit a graphical solution of all problems of feed-heater proportions for a given type of heater.

In any case the quantity of feed to be heated, the amount of auxiliary steam and its pressure and quality and the hotwell temperature will be known, and hence the problem resolves itself to finding the heater best suited to the conditions together with the final feed temperature resulting from the design. Obviously the most efficient heater is one which absorbs all of the latent heat in the auxiliary exhaust and sends it as water to the hotwell without permitting the auxiliary condenser to extract useful feed heat. Hence the available latent heat in auxiliary steam can be entirely converted into sensible heat of feed. As a rule auxiliary exhaust will carry about 15 per cent. moisture leaving 85 per cent. as dry steam. The relation between heat in steam and in feed will be given by

$$.85\ S \times r = W\ (t_o - t_i),$$

where

S = weight of auxiliary steam;

r = heat of vaporization of steam at auxiliary exhaust pressure;

W = weight of feed, all purposes;

t_o = temperature of feed leaving heater;

t_i = temperature of feed entering heater.

For convenience in calculations W is given as thousands of pounds of feed, whence S is pounds of auxiliary steam per 1,000 pounds of feed, and the above relation may be expressed

$$\frac{850\ Sr}{W} = (t_o - t_i).$$

Above a minimum value of t_i, which in practice will be 75 degrees F., the dry auxiliary steam per 1,000 pounds of feed

will be directly proportional to temperature, t_0, as shown by the straight line on figure 16, where dry steam per 1,000 pounds of feed is plotted as the ordinate with temperature as the corresponding abscissae. The second curve, length of tube against temperature of feed, is the temperature gradient as derived from experiment according to the tests described in section IV. This curve gives for one rate of flow and for the heater in question the temperature at any distance along the equivalent straight tube when the inlet water temperature has the minimum value of 75 degrees. The final curve, friction loss plotted against length of tube, measures the drop corresponding to any one rate of flow. The term "unit tube" is used because the unit is taken as the length of tube in one pass of the heater tested.

Thus there result three curves for each rate of flow and each steam pressure giving all the heaters which will absorb the entire latent heat in the auxiliary steam available. Figures 16 and 17 give the data of Tables IV and V in the form of design curves for the Bureau heater with and without retarders. To illustrate the use of these curves let it be required to design one of two feed heaters of the Bureau type without retarders for a ship using 420,000 pounds of steam per hour, all purposes, of which 12 per cent. is auxiliary steam at 10 pounds pressure.

Dry auxiliary steam per hour $= .12 \times .85 \times 420{,}000 =$ 42,840 pounds.

Dry auxiliary steam per heater per hour $= 21{,}420$ pounds.

Main feed per heater per hour $= 210{,}000$.

Dry steam per 1,000 pounds of feed $= \dfrac{21{,}420}{210} = 102$.

Hotwell temperature $= 92$ degrees.

Feed per tube per hour $= 3{,}000$ pounds.

Number of tubes required $= \dfrac{210{,}000}{3{,}000} = 70$.

Since the curves are based on a minimum temperature of 75 degrees, it is necessary to get the following data corresponding to 92 degrees:

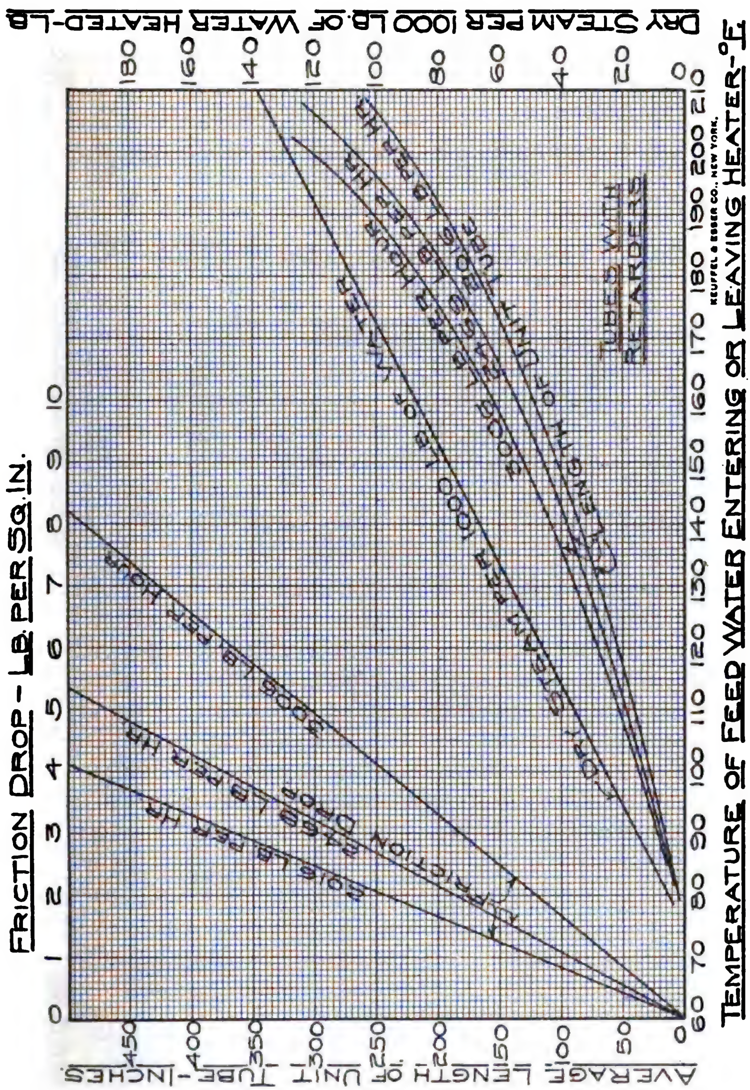


FIG. 16.

Length of unit tube, inches 32
Dry steam per 1,000 pounds feed, pounds 18

Hence

Total dry steam per 1,000 pounds of feed = 18 + 102 = 120.

Corresponding to 120 pounds of dry steam the curve gives:

Length of unit tube, inches	317.
Feed temperature, degrees	186.5
Net length of tube, inches	285.
Friction loss, pounds	5.27

Thus the heater obtained will have the sizes and give the results below:

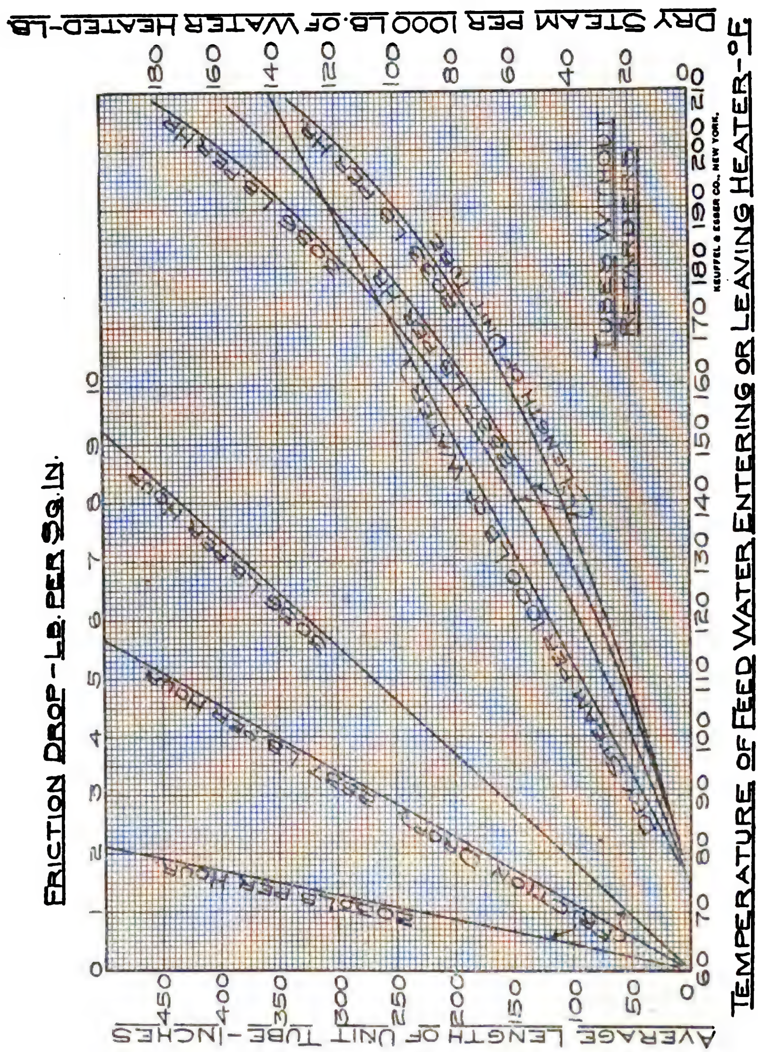
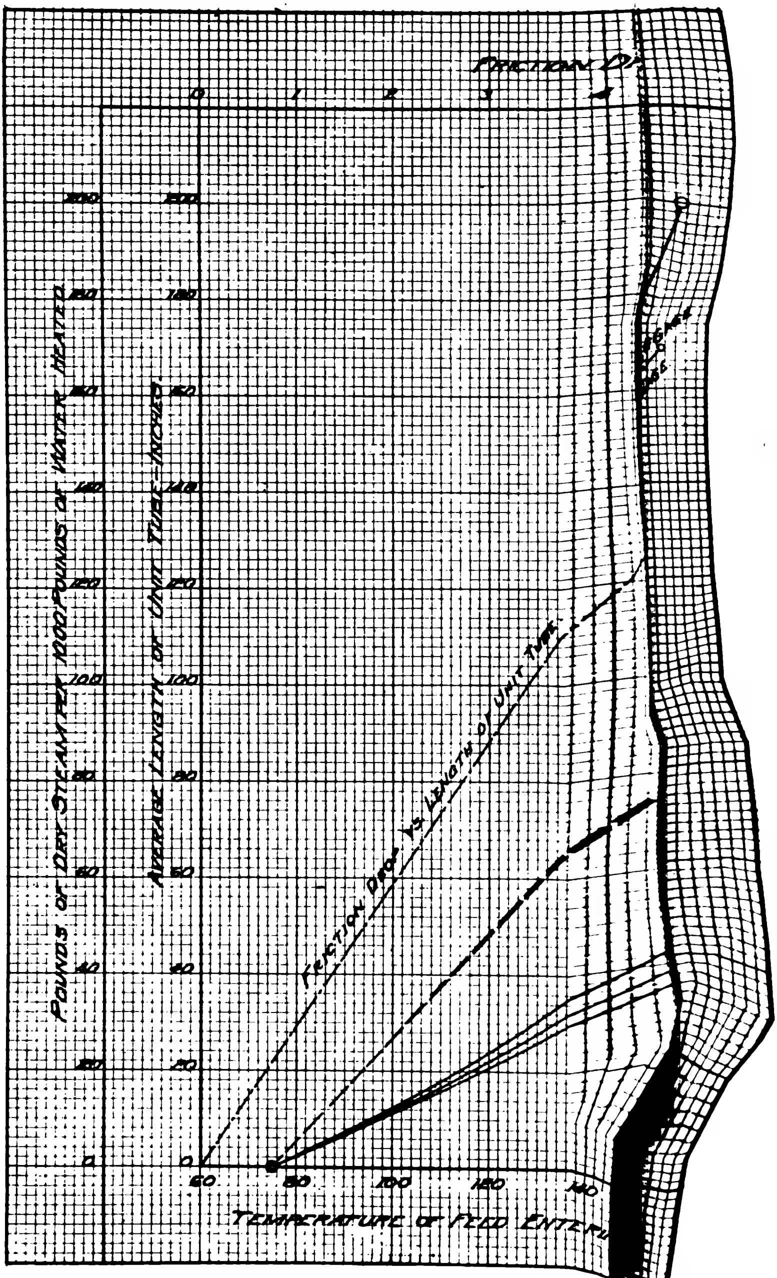


FIG. 17.



Design Curves, Showing Effect of Varying Stea Md.

Number of tubes per pass	70.
Total length of unit tube, inches	285.
*Length of unit tube per pass, inches	47.5
Number of passes	6.
Inlet-water temperature, degrees	92.
Final feed temperature, degrees	186.5
Friction loss, pounds	5.27
Heating surface per heater, square feet	334.

*Assumed as unit nearest 45.

This is a practical heater, inasmuch as the friction loss is less than 10 pounds for the maximum rate of flow. At rates of flow corresponding to 1,500, 2,000 and 2,500 pounds per tube, or 210,000, 280,000 and 350,000 pounds of feed, all purposes, the feed temperature, if the exhaust pressure remain constant, at 10 pounds will be 210 degrees, 201.5 degrees, and 190 degrees respectively.

Based on the data of Tables IX, X and XI, design curves for the Schutte and Koerting feed heater have been derived and are presented in figures 18, 19 and 20 for water rates of 5,000, 8,000 and 10,000 pounds per tube per hour and steam pressures of 5, 10 and 15 pounds. These curves are likewise based on a minimum hotwell temperature of 75 degrees F. A water rate of 5,000 pounds per tube is the equivalent of builder's rated capacity of the heater tested, but the test results indicated that full economy of heating surface was not realized at that rate. The higher values of 8,000 and 10,000 pounds were selected because they fall about evenly on the two sides of the present allowable friction drop of 10 pounds for a two-pass heater. The mean length of tube in each of two passes is 51⅝ inches, and the heating surface for a pair of tubes is 5.11 square feet; so the equivalent heating surface for lengths of 50, 100, 150 and 200 inches would be 4.95, 9.90, 14.85 and 19.80 square feet.

The design curves in figures 18 and 19 show the variation of outlet temperature, dry steam per thousand pounds of water and friction drop with length of tube and with steam pressure when the rate of flow in pounds per tube is maintained constant, whereas the curves in figure 20 show the variation of

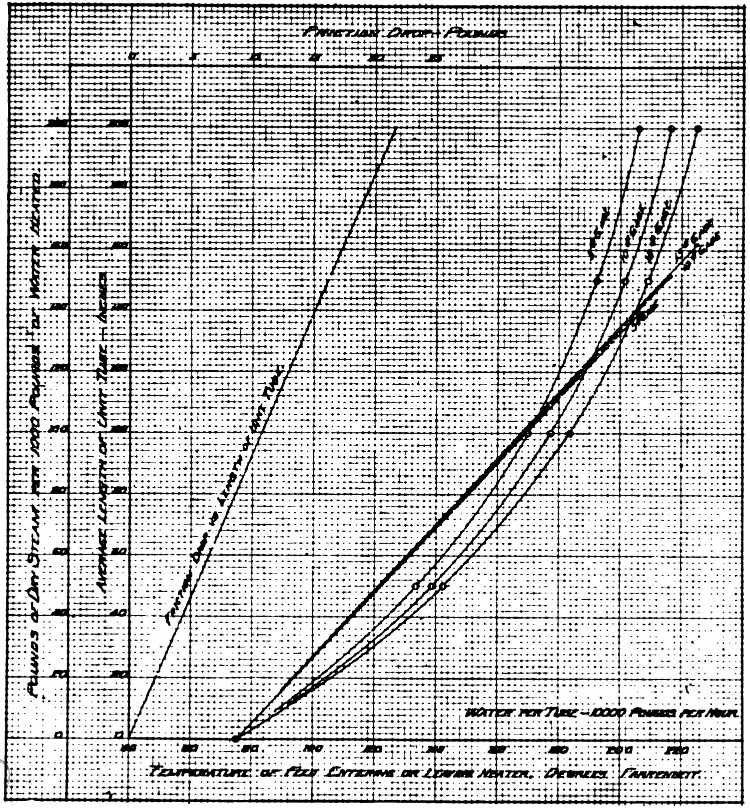


FIG. 19.—DESIGN CURVES SHOWING EFFECT OF VARYING STEAM PRESSURE. SCHUTTE-KOERTING FEED-WATER HEATER WITH 3/16-INCH FILM. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

——— Average length of unit tube.
— — — Pounds of dry steam per 1,000 pounds of water heated.

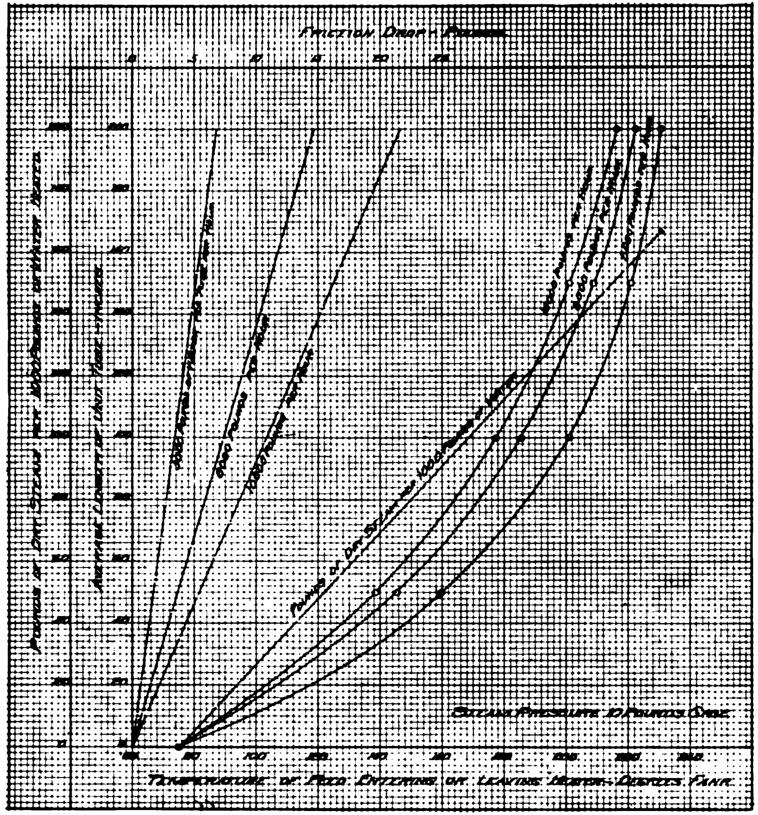


FIG. 20.—DESIGN CURVES SHOWING EFFECT OF VARYING WATER RATE. SCHUTTE-KOERTING FEED-WATER HEATER WITH 3/16-INCH FILM. U. S. NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MD.

——— Average length of unit tube.
—— —— Pounds of dry steam per 1,000 pounds of water heated.
—— . —— Friction drop *vs.* length of unit tube.

the several quantities with rate of flow when steam pressure is maintained constant at 10 pounds gage.

VII.—Effect of Operating Conditions on Feed Temperature.

At a water rate of 8,000 pounds per tube per hour the length of tube required to heat water from 90 degrees to 210 degrees with exhaust steam at 10 pounds gage is 144 inches. Reducing the steam pressure to 5 pounds with the same tube would yield an outlet temperature of 200 degrees, whereas increasing the pressure to 15 pounds would increase the outlet to 218 degrees F. A decrease of 10 degrees in feed temperature represents a loss of 10 B.t.u. per pound of steam generated, or 1 per cent. of the heat absorbed in the boilers, whereas decreasing back pressure from 10 to 5 pounds would effect a reduction in steam consumption of turbine-driven auxiliaries from 43.9 to 40.2 pounds per shaft horsepower per hour,* a saving of 8½ per cent. in this item of auxiliary steam. Thus where auxiliary steam is in excess of 12 per cent. of total steam it appears to be in the interest of overall economy to obtain as low an auxiliary exhaust pressure as possible and sacrifice high feed temperature.

Considering the effect of variation in water rate at a constant steam pressure, the curves show the length of tube required to heat water from 90 degrees to 210 degrees at 10,000 pounds per tube to be 166 inches with a friction drop of 17.9 pounds. If the heater be designed for these conditions for maximum load, then the outlet temperatures for 8,000 and 5,000 pounds per tube would be 217 degrees and 226 degrees with corresponding friction drops of 12.2 and 5.7 pounds.

The economy of increased water rate may best be shown by calculations for one of two heaters for a ship whose steam consumption for all purposes is 480,000 pounds per hour of

*See report of test of Worthington centrifugal feed pump driven by Terry turbine, Journal American Society of Naval Engineers. Assumed mechanical efficiency of pumps is 50 per cent.

which 60,000 pounds is used by auxiliaries. The results are as follows:

Total water fed, pounds per hour			480,000.
Auxiliary steam, pounds per hour			60,000.
Quality of auxiliary exhaust			.85
Water per heater, pounds per hour			240,000.
Dry auxiliary steam per heater, pounds per hour			25,500.
Temperature of inlet to heater, degrees			90.
Water, per tube	5,000	8,000	10,000
Number of tubes	48	30	24
Dry steam per 1,000 lbs. of feed per hour	106	106	106
Add for increased temperature of inlet water	16	16	16
Total steam per 1,000 lbs. of feed	122	122	122
Corresponding outlet temperature	189	189	189
Length of tube, inches	82	107	122
Subtract for 90 degrees inlet water	6	9	10
Net length of tube, inches	76	98	112
Friction drop, pounds	2.6	7.2	11.0
Heating surface per inch, square foot	.099	.099	.099
Total heating surface per unit	361.	291.	266.

Selecting the 112-inch heater with 24 units would produce results at 50 per cent. and 80 per cent. capacity of

Water, per tube, pounds	5,000.	8,000.
Water, per heater, pounds	120,000.	192,000.
Outlet feed temperature with 10 lbs. auxiliary exhaust	211.5	197.
Outlet feed temperature with 5 lbs. auxiliary exhaust	201.5	189.
Friction drop, pounds	3.5	7.5

In each case at reduced power there would be excess auxiliary steam to be bled to the auxiliary condenser, whereas at full power the heat absorbed by the feed heaters should care for the entire condensation.

VIII.—Air in Feed Heaters.

In the tests previously noted precautions were taken to vent all accumulated air from the steam side of the heaters. Water will absorb nearly 2 per cent. by volume of air and this air will

be carried over by the steam. As has been pointed out, at a given pressure and temperature of saturation air has a greater density than the steam with which it is associated, and it will gradually accumulate above the condensation in the bottom of the heater. Thus part of the surface is rendered inactive and the film resistance is increased to an extent that heat transfer is appreciably reduced. This blanketing effect will be greater in a film heater with horizontal tubes than if the tubes be vertical, and in the latter case the usual precautions of blowing through pet cocks in the covers would be ineffective.

The extent to which accumulated air will reduce heat transfer is to be noted from Table XII, which contains data from a two and one-half hour test on the film heater, during which time no air was relieved from the steam space and the water seal permitted only the release of water. The rate of heat transmission decreased steadily from 661 B.t.u. per square foot per degree average temperature difference to 558 B.t.u., a net loss of 15.6 per cent. At the same time the hot feed temperature fell from 195.1 degrees to 181.8 degrees. Although full steam pressure of 10 pounds was maintained, the temperature at the bottom of the heater fell from 234.7 degrees to 216.5 degrees, and two lower temperatures were noted during the test. Automatic air valves are not suitable for this service, and the best remedy is pet-cock relief with a moderate but continuous blow of steam from the bottom of the steam space.

IX.—Auxiliary Steam for Feed Heating.

The amount of auxiliary steam available for feed heating depends on the type and size of vessel and the character of the main and auxiliary equipment. From the trial trips which have been reported recently in the Journal of the American Society of Naval Engineers the data of Table XIII has been collected, incorporating the steam consumption of such vessels where the main-engine steam was separated from the auxiliary steam.

				Water Data.									Steam Data.							Deduced Data.				
Number of Observation.	Date of Run. October, 1912.	Time of Observation.	Room Temperature.	River Water Pumped Through Heater, Pounds per Hour.	Temperature at Inlet, Degrees F.	Temperature at Outlet, Degrees F.	Temperature Rise, Degrees F.	Specific Heat of Water.	B.T.U. Absorbed by Water per Hour.	Friction Drop, Inches of Mercury.	Water Pressure, Outlet.	Friction Drop, Pounds per Sq. In.	Pressure by Gage at Inlet.	Pressure, Pounds Absolute, at Inlet.	Temperature at Inlet, Degrees F.	Temperature Corresponding to Pressure.	Degrees Superheat.	Temperature at Outlet of Heater, Degrees F.	Temperature of Condensed Steam.	B.T.U. Transmitted per Sq.Ft. per Hour.	Average Arithmetical Temperature Difference.	B.T.U. Absorbed per Sq.Ft. per Hour per Degree Average Temp. Diff.	Water Heated per Sq.Ft. of Heating Surface, Pounds per Hour.	Water per Tube, Pounds per Hour.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
1	25	1:30	75	———	61.9	195.1	133.2	.974	———	24.0	183	10.9	10.3	25.0	301.6	240.1	61.5	234.7	234.2	———	111.6	———	———	———
2	25	1:40	76	12151	61.9	194.4	132.5	.974	1568700	22.3	183	10.2	10.0	24.7	293.8	239.4	54.4	231.8	230.9	73540	111.2	661	570	6076
3	25	1:50	77	12112	61.8	193.0	131.2	.974	1547900	21.8	187	9.9	9.8	24.5	291.0	239.0	52.0	228.5	228.1	72570	111.6	650	568	6056
4	25	2:00	75	11719	61.8	189.7	127.9	.974	1460200	21.0	183	9.6	9.9	24.6	287.9	239.2	48.7	228.0	225.3	68450	113.4	604	549	5860
5	25	2:10	72	11719	61.8	194.2	132.4	.974	1511800	21.0	188	9.6	10.0	24.7	273.5	239.4	34.1	225.4	224.7	70870	111.4	636	549	5860
6	25	2:20	73	11680	61.9	191.7	129.8	.974	1476400	20.0	185	9.1	10.0	24.7	266.4	239.4	27.0	222.3	222.1	69210	112.6	615	548	5840
7	25	2:30	73	11837	62.0	199.0	137.0	.974	1579100	22.8	180	10.4	10.0	24.7	266.1	239.4	26.7	222.7	222.3	74030	108.9	680	555	5919
8	25	2:50	73	13095	61.9	192.4	130.5	.974	1664400	24.8	229	11.3	10.0	24.7	266.5	239.4	27.1	218.7	219.9	78030	112.2	695	614	6548
9	25	3:00	74	12899	62.0	192.9	130.9	.974	1644600	22.1	226	10.1	10.0	24.7	272.0	239.4	32.6	221.4	220.1	77100	111.9	689	605	6450
10	25	3:10	73	11798	62.1	189.1	127.0	.974	1459400	22.4	236	10.2	10.0	24.7	267.6	239.4	28.2	220.0	218.9	68420	113.8	601	553	5899
11	25	3:20	73	11994	62.1	189.9	127.8	.974	1493300	22.2	236	10.1	10.0	24.7	267.9	239.4	28.5	218.5	217.9	70000	113.4	617	562	5997
12	25	3:30	74	11915	62.2	189.3	127.1	.974	1475100	21.8	238	9.9	9.9	24.6	269.2	239.2	30.0	217.7	216.9	69150	113.4	610	559	5958
13	25	3:40	74	11798	62.2	189.9	127.7	.974	1467700	21.6	236	9.8	10.1	24.8	264.1	239.6	24.5	212.5	212.3	68810	113.5	606	553	5899
14	25	3:50	74	11758	62.2	186.2	124.0	.974	1420400	21.5	231	9.8	10.0	24.7	270.4	239.4	31.0	207.7	208.3	66590	115.2	578	551	5879
15	25	4:00	74	11994	62.2	181.8	119.6	.974	1397300	22.4	231	10.2	10.0	24.7	241.4	239.4	2.0	216.5	216.1	65510	117.4	558	562	5997

TABLE XII.

TEST OF SCHUTTE-KOERTING EXPERIMENTAL FEED-WATER HEATER. 7/64-INCH WATER FILM.—VERTICAL ARRANGEMENT.

TABLE XIII.—MAIN AND AUXILIARY STEAM CONSUMPTION OF NAVAL VESSELS.

Vessel and type.	Speed, knots.	Steam per hour, pounds.		Auxiliary steam, per cent. of total.	Auxiliary exhaust pressure.	Feed temperature.	
		Auxiliaries.	All purposes.			At hot well.	At boilers.
Delaware	21.56	40,557	422,931	9.6	...	104.5	...
Engined battleship.	19.22	30,512	243,810	12.5	...	89.7	...
	12.24	23,457	83,462	28.1	...	90.7	...
Utah	21.04	62,400	406,397	15.4	...	90.1	173.1
Turbined battleship.	19.23	47,400	262,498	18.1	...	91.5	177.5
	12.02	32,200	98,498	32.7	...	93.0	213.0
Florida	22.08	67,352	560,880	12.0	6.7	74.0	197.8
Turbined battleship.	19.19	56,382	308,051	18.0	4.9	81.9	201.0
	12.08	41,843	116,386	35.9	6.4	79.5	223.3
Birmingham	10.03	12,081	34,069	35.2	...	...	...
Engined scout cruiser.	14.95	15,657	65,412	33.9	...	60.0	146.4
	19.86	22,217	132,037	16.8	...	...	...
	22.55	29,684	214,894	13.8	...	...	...
	24.15	43,928	297,300	14.8	...	...	...
Salem	10.20	13,446	46,598	28.8	...	...	...
Turbined scout cruiser.	14.85	17,967	82,947	21.7	...	73.5	172.9
	20.16	20,337	157,519	12.9	...	...	...
	22.55	33,520	224,826	14.9	...	...	...
Chester	10.15	14,151	47,690	29.6	...	...	...
Turbined scout cruiser.	15.81	18,369	95,621	19.2	...	98.0	212.2
	20.05	25,569	158,220	16.1	...	...	...
	22.78	29,002	205,553	14.1	...	...	...
	24.67	42,392	296,054	14.3	...	...	...
Parker and *Benham*	29.57	36,146	283,633	12.7	...	56.9—59.4	183.3—193.3
Destroyers, turbines and cruising engines.	24.06	18,066	138,491	13.0	...	53.3—55.3	189.6—209.3
	15.56	5,523	32,281	17.1	...	46.1—51.8	210.3—214.4
	12.00	5,253	19,717	26.6	...	46.0—53.6	199.5—221.6
Paulding	32.80	33,000	218,655	15.1	...	96.4	187.2
Turbined destroyer.	25.31	23,000	105,630	21.8	...	88.0	221.0
	15.86	12,500	30,967	40.4	...	71.0	222.0
New York	21.47	52,336	495,183	10.56	5.3	94.0	190.6
Engined battleship.	19.23	53,296	300,038	17.78	11.8	92.8	216.8
	12.11	36,236	102,645	35.42	14.9	76.7	236.2

Where steam-consumption guarantees of machinery for proposed installations are not available, this table may with safety be used for a vessel of similar class and equipment.

X.—CONCLUSIONS.

From the tests considered herein the following summary applies:

(1) Closed feed-water heaters developed for marine purposes have temperature gradients resulting from the general

relation that heat transfer is proportional to some power of the temperature difference;

(2) The numerical value of the exponent for the heaters tested is less than unity;

(3) A vertical tube permits higher rates of heat transfer than a horizontal tube due to the more effective disposal of condensate;

(4) Film-tube heaters give much higher rates of heat transfer than plain tubes or tubes with retarders;

(5) The clearance between the tubes making up a film unit can be great enough to prevent excessive friction loss in the form of pressure drop without in the slightest decreasing the efficiency of the film agitation, which renders this type more effective for heat transfer than a plain tube;

(6) Air carried into the steam space of a feed-water heater will seriously interfere with heat transmission in any type of heater if allowed to accumulate for any considerable period of time;

(7) Water-film agitation is more important than steam side-film agitation;

(8) Water-film agitation and air-free condensation are essential to the maintenance of high rates of heat transfer.

APPENDIX I.

Principal Dimensions of Schutte and Koerting Film-Oil Heater Illustrated in Figure 8.

The following dimensions were obtained by measurement after the completion of the test:

Weight, empty, pounds	507½
Length over all, feet and inches	6–05
Width over external steam connections, inches	24
External diameter, inches	11⅛
Outer tube:	
Inside diameter, inches	4.03
Depth of corrugations, inch	0.62
Pitch of corrugations, inches	2
Thickness (assumed), B.W.G.	9
Length between headers, inches	43½

Inner tube:	
Outside diameter, inches	5.04
Depth of corrugations, inch	0.68
Thickness, inch	0.11
Length between heads, inches	54
Steam connections, inch	1
Oil connections, inches	1½
Distance between center lines of oil connections, inches	41½

The following values have been calculated from the above dimensions:

Outer surface of outer tube, square feet	5.58
Inner surface of inner tube, square feet	6.13
Total heating surface, square feet	11.71
Maximum inside diameter of outer tube, inches	5.27
Maximum outside diameter of inner tube, inches	5.04
Clearance, inch	0.115
Sectional area of water film, square inches	1.86
Minimum inside diameter of outer tube, inches	4.03
Minimum outside diameter of inner tube, inches	3.68
Clearance, inch	0.175
Sectional area of water film, square inches	2.12
Average sectional area of water film, square inches	1.99

APPENDIX II.

Principal Dimensions of Schutte and Koerting Experimental Feed-Water Heater Illustrated in Plate II.

The principal dimensions of the apparatus are:

Length over all, feet and inches	6–09¼
External diameter of shell, inches	10¾
Internal diameter of shell, inches	10¼
Face to face of water connections, inches	18¼
Outer tubes:	
Length over all, feet and inches	3–11
Maximum inside diameter, inches	2.373
Minimum inside diameter, inches	1.813
Maximum outside diameter, inches	2.563
Minimum outside diameter, inches	2.003
Depth of corrugations, inch	.280
Pitch of corrugations, inches	1.125
Thickness, B.W.G. number	13
Thickness, inch	.095
Inner tubes, 7/64-inch film:	
Length over all, feet and inches	4–10¼
Maximum inside diameter, inches	1.997
Minimum inside diameter, inches	1.375
Maximum outside diameter, inches	2.163
Minimum outside diameter, inches	1.541

Inner tubes, 7/64-inch film:

Depth of corrugations, inch	.311
Pitch of corrugations, inches	1.125
Thickness, B.W.G., number	14
Thickness, inch	.083

The 3/16-inch film is constructed by using the same outer tube as in the 7/64-inch film, and making the inner tube from the same blank as the inner tube with smaller film, but driving the forming tool into the blank tube to a greater depth.

The resulting dimensions for the inner tubes for 3/16-inch film would be:

Length over all, feet and inches	4–10¼
Maximum inside diameter, inches	1.936
Minimum inside diameter, inches	1.193
Maximum outside diameter, inches	2.102
Minimum outside diameter, inches	1.359
Depth of corrugations, inch	.311
Pitch of corrugations, inches	1.125
Thickness, B.W.G. number	14
Thickness, inch	.083

From the dimensions given in the first paragraph and radii of arcs of corrugations shown on Plate II the following values have been calculated:

Nominal film thickness, inch	$0.\frac{7}{64}$
Actual film thickness, inch	0.120
Outer surface of one outer tube, square feet	5.155
Inner surface of one inner tube, square feet	5.510
Total heating surface in tubes, square feet	21.33
Sectional area of each water film, normal to axis of tube, square inch	0.744
Total area of water flow, normal to axis of tube, square inches	1.488

Similarly there have been derived for the 3/16-inch film the following:

Nominal film thickness, inch	$0.\frac{3}{16}$
Actual film thickness, inch	0.1875
Outer surface of one outer tube, square feet	5.155
Inner surface of one inner tube, square feet	5.070
Total heating surface in tubes, square feet	20.45
Sectional area of each water film, normal to axis of tube, square inches	1.145
Total area of water flow, normal to axis of tube, square inches	2.290

APPENDIX III.

Method of Obtaining Items in Tables VII to XI. Test of Schutte and Koerting Feed-Water Heater.

Item,

1, 2. For reference.

3. Observed.

4. From data corrected for temperature according to Smithsonian Meteorological Tables.

5. Observed.
6. From data obtained from calibrated water meter.
7, 8. Observed.
9. Item 8 — item 7.
10. From chemist's determination.
11. From Smithsonian Physical Tables corresponding to density in item 10.
12. Item 6 × item 9 × item 11.
13. From observation of differential mercury gage.
14, 15. Observed.
16. Where friction drop was within the limit of the 40-inch differential gage used this item was calculated from item 13 × 0.4554, where 0.4554 is the decimal of a pound per square inch equivalent to 1 inch elevation of a mercury and water differential gage. Otherwise, friction drop is measured by item 14 — item 15.
17. Observed.
18. Item 17 + (0.491 × item 4).
    0.491 inches of mercury = 1 pound per square inch.
19. Observed.
20. From steam tables corresponding to item 18.
21. Item 19 — item 20.
22, 23. Observed.
24. $Q = H - q$,

where

Q = B.t.u. taken from each pound of steam.

H = Heat content of one pound of steam corresponding to conditions of items 18 and 21.

q = Heat of the liquid corresponding to item 23.

25. Observed.
26. Item 24 × item 25.
27. Item 26 — item 12.
28. From formula $W' = W \frac{H - q}{H' - q}$.

where

W' = Equivalent dry steam,

W = Actual steam per hour = item 25,

H = Total heat in one pound of inlet steam corresponding to items 18 and 21,

H' = Total heat in one pound of dry, saturated steam corresponding to item 18,

q = Heat of the liquid corresponding to item 23.

29. $$V = \frac{W}{D \times \frac{A}{144} \times 60},$$

where

V = Axial velocity in feet per minute,

W = Weight of feed water per hour = item 6,

A = Area in square inches of 2 film spaces normal to axis of tube = 1.488 square inches for 7/64-inch film and 2.290 square inches for 3/16-inch film.

D = Density of river water in pounds per cubic foot corresponding to items 7 and 10.

30. $\frac{\text{Item 12}}{\text{A}}$,

where A is the area in square feet of water-heating surface = 21.33 square feet for 7/64-inch film and 20.45 square feet for the 3/16-inch film.

31. $t = t_s - \frac{t_1 + t_o}{2}$

t = Average arithmetical temperature difference,
t_s = Temperature of saturated steam = item 20,
t_1 = Temperature of inlet water = item 7,
t_o = Temperature of outlet water = item 8.

32. $t_m = \dfrac{t_o - t_1}{\text{Log}_e \left(\dfrac{t_s - t_1}{t_s - t_o}\right)}$.

t_m = Mean geometrical temperature difference,
t_s, t_o and t_1 same as in item 30.

33. $\frac{\text{Item 30}}{\text{Item 31}}$.

34. $\frac{\text{Item 30}}{\text{Item 32}}$.

35. $\frac{\text{Item 28}}{\text{Item 6}} \times 1{,}000$.

36. $\frac{\text{Item 6}}{\text{A}}$,

where A is the water heating surface in square feet as given in item 30 above.

37. $\frac{\text{Item 12}}{\text{Item 26}}$.

38. $\frac{\text{Item 6}}{2}$.

STEAM TURBINE BLADE FASTENINGS.

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So far as we are aware the era of the steam turbine dates back to 120 B. C., when Hero, of Alexandria, constructed his reaction wheel. Little is known of the use to which his turbine was put, but we can be pretty safe in saying that it amounted to little more than a philosopher's toy.

About the year 1630 A. D. Branca, an Italian, conceived the idea of utilizing the kinetic energy of steam and built a pure impulse turbine with a view to supplying power to take the place of that of man and beast.

The nucleus of the great prime mover of today lay dormant until the 19th century, when the possibilities of a velocity engine again revealed itself and engineers were beginning to appreciate the value of a rotary steam motor.

Wm. Gilman and others tackled the subject, but their lack of knowledge of the principles of steam in motion left their turbines little further than in their experimental stages.

Sir Charles A. Parsons' name stands out before us as the real pioneer of the steam turbine. His first patents were filed in the year 1884. Claims on blade fastenings have occupied no mean place in the patent records since that time, and it is with the most important of these I propose to deal in this paper.

All along the line turbine engineers have been striving to increase the efficiency of their engines. Repeated tests have proven that the efficiency varies with the blade speed and the number of rows of blades, and that the number of rows of blades varies inversely as the square of the blade speed for a

given efficiency. That is to say, if a turbine running at 600 revolutions per minute requires 64 rows of blades to show a certain efficiency, another turbine having the same mean blade diameter and the same height of blades, would only require 8 rows of blades, if running at 1,200 revolutions per minute for the same efficiency. From this it will be seen that increasing the number of revolutions per minute offers increased efficiency with smaller turbines, and the tendency is to develop along these lines.

The adoption of reduction gearing opens up a bright future for the further development of the steam turbine. With the direct drive the revolutions of the turbine have to be kept low for propeller considerations; but with gearing the turbine can be designed as an independent unit, without regard to the propeller revolutions. The question is how far can we go on increasing the peripheral speed of the blades and reducing the size of the turbine. For the answer we must look to the strength of the materials used in their manufacture. Theoretically the limit would be reached when the mean blade speed equals between .75 and .9, that of the steam in the case of a Parsons reaction turbine, and .5 in the case of a pure impulse turbine. Quite a common pressure drop through a turbine is from 265 pounds per square inch down to 28 inches vacuum. This drop in pressure represents a steam velocity of 4,200 feet per second.

The possibility of ever reaching the theoretical blade limit is indeed remote, but the tendency is to go on increasing the blade speed. It will be of interest to note here that Messrs. Parsons seldom exceed 550 feet per second at the mean diameter of the last expansion; Curtis as high as 700 feet per second, and DeLaval 1,375 feet per second; all a long way behind the theoretical most efficient speed. With the increase of the revolutions if the diameter is kept down the blade heights must run up. This cannot be done without taxing the strength of the blade and its fastenings.

In the turbines of today it is highly essential to bind a number of the blades in a row together, either by means of a shroud-

ing or lacing wires, in order to strengthen them against the fatiguing effect of the high period of vibration set up in the blade when running.

It would, of course, be desirable to have this shrouding or lacing, as the case may be, continuous, but, owing to expansion troubles due to the use of different metals for the turbine rotor and the blades, this is impracticable. Provision must be made to take care of the expansion. Previous to the adoption of the blade ties, blading gave no end of trouble through breaking at the roots. In passing I might mention that shrouding is used on impulse turbines to serve a double purpose. Besides tying the blades it prevents the lateral spreading of the steam jet on its passage across the turbine blade. Shrouding is not absolutely essential on reaction turbines, as the radial clearances between the rotor-blade tips and the cylinder walls are very fine; but one form of binding must be used. Channel shrouding is used by some builders as a safeguard against damage through-blade stripping. At the time of its conception blade stripping was of common occurrence; if one single rotor blade fouled with the cylinder wall the destruction of at least the rows in which it was contained was almost certain.

For some years now Parsons have adopted the clever idea of thin tipping their turbine blades for a small portion of their length, and should a revolving blade creep far enough from its root fastening to foul the casing the sharpened tip of the blade will simply be turned over and the rest of the row will not be affected.

In recent years impulse turbines have come to the front. The writer believes that they have much to recommend them, and when the mechanical design receives the attention it deserves, more will be heard of them in naval work. The fact that Messrs. Parsons have incorporated an impulse element in their reaction turbine lends some color to this remark.

With the development of the impulse turbine in particular the tendency to speed up is most marked, and this is bringing into being blades of much heavier section than have hitherto been used. Heavier blades and higher velocities keep the de-

signer's mind constantly working in the endeavor to provide a root fastening strong enough and reliable enough to take care of the increasing loads imposed upon it. That this is no easy proposition must be admitted.

Perhaps no field in the design of the steam turbine has been so much exploited as that of blade fastenings, as a survey of the patent records will show. Many ingenious methods have been devised from time to time, the most important of which will be detailed later. Some of the fastenings used today, though highly suitable to meet the conditions existing at the time they were devised, must soon be consigned to the scrap heap or museum, and give place to a fastening of unquestionably superior strength. Too much stress cannot be laid upon the importance of strength. No blade fastening can be too strong. Most builders have, at some time or other, made up a test piece of their particular form of blade fastening and had it pulled to destruction in a testing machine, and demonstrated that it required a pull far exceeding the centrifugal force to which it would be subject when running to destroy it. But, while a carefully-prepared test piece pulled in a testing machine affords excellent material for comparison with other methods, it is poor practice to use the test results as a criterion for running conditions. The unknown quantity enters too largely into this equation. To repeat myself, "The direct pulling strength of a blade fastening ought to be as great as possible, but not only that, the blade must be rigidly held in every direction with no possible clearance to allow of movement."

Perfect rigidity is the hall-mark of excellence in a blade fastening, especially so in an axial direction, for upon this, to a very large extent, depends the life and reliability of the fastening. If there should be any initial clearance between the blades and the sides of the grooves the vibration to which the blades are subject when running under steam may be the means of working the blades loose. Machine fits cannot be relied upon to ensure the desired side fit, and caulking must be resorted to. Without caulking we have no guarantee that the

blades completely fill the width of the groove. Caulking is the best known method of assuring biting contact; but while the writer is a keen advocate of its use to ensure rigidity he is not in favor of a blade fastening which relies upon caulking for its direct pulling strength. Caulking should be employed only as a means of eliminating freedom of movement. Heavy caulking is to be avoided on turbine rotors and discs. It distorts them and calls for increased pitches of rows and heavier rotors and discs. In high-speed turbines it is desirable, for reasons of efficiency, to keep down the axial pitch of the rows as much as possible. To do so with a blade fastening of questionable strength and rigidity would be courting disaster.

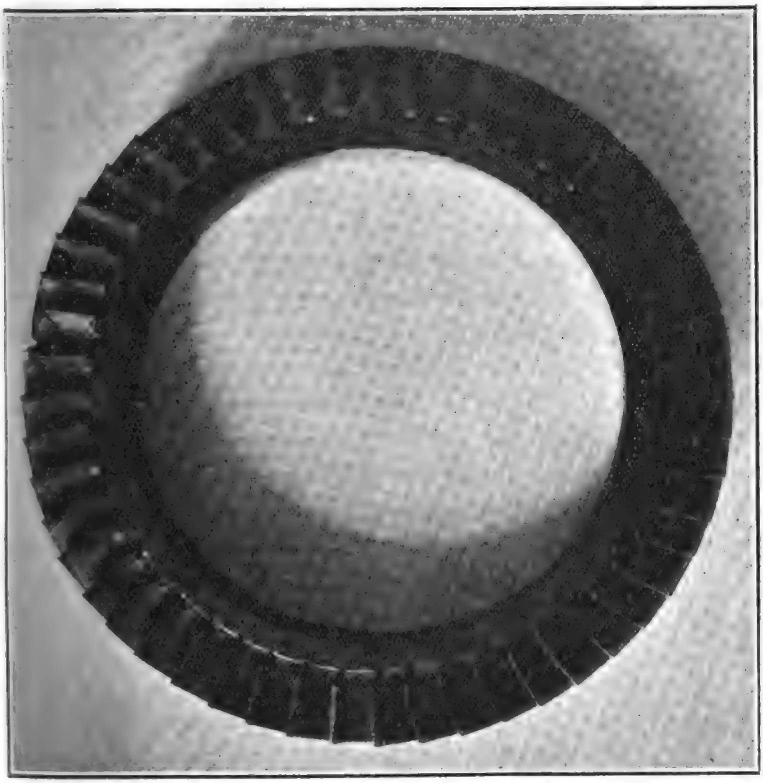


FIG. 1.

Before proceeding with a general description of the various methods of blade fastenings, mention should be made of the desirability of giving liberal depth to the groove, as the blade has to rely upon the depth of its fastening for its steadyment.

The blades of the earliest machines built by Messrs. Parsons were made in complete rings which were keyed onto the turbine shaft. The blades were formed by saw cutting the outer circumference of the rings and twisting the edges with pinchers to the desired angles. Experience went to show that while this method might provide the correct angles of entrance and discharge, it left much to be desired in the way of the gradual changes of the course of the steam on its passage across the blade and the smooth polished surface so essential to the efficiency of the turbine. To embody these requirements in a blade was a problem which was solved by making the blades in lengths and sawing to size. Next came the all-important question as to the best means of securing these innumerable blades to their revolving elements. The idea of grooving the rotor, inserting the blades and caulking the distance pieces between, proved to be a good line along which to work. This method is still used by Messrs. Parsons, although the grooves themselves have undergone a number of changes. At first the grooves were plain dovetailed. Then came dovetailed grooves with the sides serrated with grooves of various designs and proportions passing through the rectangular, semicircular and vee shapes. After some years the grooves proper were made parallel and the vee-shaped serrations adopted.

Tests showed that the metal of the distance pieces when caulked flowed more freely into the vee serrations than the other types mentioned. The depth of the grooves vary from ¼ inch for blades ¼ inch wide, to ⅝ inch for blades 1 inch wide.

The distance pieces are cut 1/32 inch longer than the depth of the grooves to provide metal for displacing, so that after caulking, the distance pieces are flush with the rotor surface. A good idea of the proportion a Parsons standard reaction blade fastening for rotors may be seen in Fig. 2.

Fig. 3 shows the Parsons' method as applied to impulse blades. Owing to the heavier sections the rotor grooves are made deeper than those for the same width of reaction blades, and in order to ensure contact at the lower part of the root the distance pieces are double caulked, or, in other words, inserted in two pieces—the lower being caulked hard up before the top piece is inserted. The fronts and backs of the blades at the roots are serrated with vee-shaped serrations into

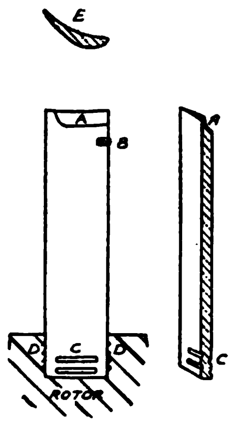


FIG. 2
PARSONS STANDARD
REACTION BLADE
'A' THIN TIPPING. 'B' LACING STRIP
'C' INDENTATIONS 'D' SERRATIONS IN ROTOR.
'E' BLADE SECTION.

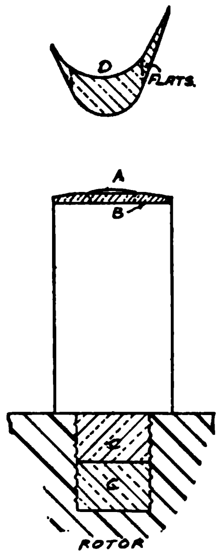


FIG. 3
PARSONS STANDARD
IMPULSE BLADE.
'A' RIVETING 'B' SHROUDING.
'C' DOUBLE CAULKED DISTANCE PIECES
BETWEEN BLADES. 'D' BLADE SECTION.

which metal of the distance piece is driven when caulked. It will be noticed in this type of fastening that the holding power of the blade is solely dependent upon its adjacent distance piece; whereas in the Parsons reaction fastening, where the blades are singly inserted, the blades contribute to the holding power, due to the fact that the rotor groove is made narrower than the width of the blades, and the blades are fullered through a degree or two in driving them together in the

groove, thereby making a biting contact between the edges of the root of the blades and the side of the rotor groove; the side flats existing at the roots after cutting back the fine edges of the blades do not permit of this being done in the case of impulse blades.

It is the practice with a good many impulse-turbine builders to allow the distance piece to project up the blade beyond the top of the rotor groove to give the blade increased strength against bending and also provide a truer steam line across the stage. With the Parsons method and other methods in which the distance pieces are used as caulking pieces, this desirable feature cannot be taken advantage of.

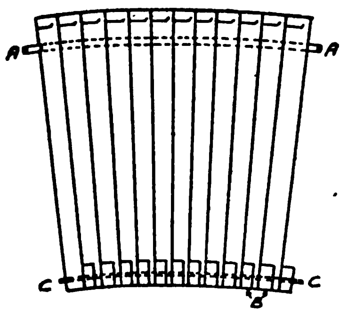


FIG. 4
SEGMENT OF PARSONS ROSARY BLADING
"A." LACING STRIP. "B." DISTANCE PIECES. "C." WIRE THROUGH ROOT.

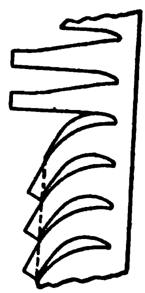


FIG. 5
PARSONS COCKSCOMB ROOT.

The Parsons method has proved itself to be an excellent fastening for reaction blades running up to 550 feet per second, and suitable for impulse turbines of low blade speed, but it has yet to prove itself capable of taking care of the higher blade speeds suggested as a means of developing maximum efficiency with minimum dimensions. You are referred to the table further on in this paper for test fixtures on the Parsons blade fastenings. In order to shorten the time of erection and facilitate repairs, Messrs. Parsons build up their blading in segments such as shown in Fig. 4.

The blades and distance pieces have a hole drilled through them near the base and are assembled in a former with a wire threaded through them, which is turned over at the ends to hold the segment together until ready to put into rotor. This type goes by the name of the Rosary Blading. An earlier attempt at building up this blading in segments is shown in Fig. 5 and was known as the cock's comb method. It derived its name from the shape of its foundation ring, which was milled out at intervals to receive the blades, giving to it a close resemblance of a long cock's comb. After the blades were inserted the points of the comb were knocked over and thus held the blades in position. The life of this type was a comparatively short one. The fact that it required a groove about 25 per cent. wider than the blade was an objectionable feature.

Segmental blading after being inserted in the rotor groove must be individually caulked. This individual caulking required of the Parsons method, constitutes, to the writer's mind, a weakness, for the reliability of the fastening is dependent upon the scrupulosity of the blader.

Other types of blade fastenings patented by Messrs. Parsons and his associates are shown in Figs. 6 and 7:

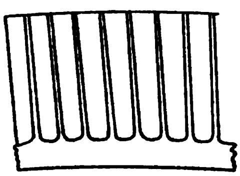


FIG. 6
PART SEGMENT OF EARLY PARSONS BLADING CUT FROM SOLID PIECE OF DELTA METAL.

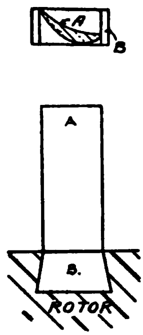


FIG. 7
PARSONS BLADE AND DISTANCE PIECE COMBINED.
'A' BLADE SECTION 'B' BASE.

In Fig. 7, the blades and distance piece are in one and are let into a dovetailed groove, and the distance-piece projection is caulked in the usual Parsons method.

An ingenious blade fastening is that of Ferranti. In its simplest form the blades are made of nickel-coated mild steel and electrically welded to the turbine rotor. For marine work the blades are welded to mild-steel foundation rings, which are secured to the rotors; the object being to enable spares to be carried and fitted easily. Such an arrangement is shown in Fig. 8. The rotor or foundation ring is serrated and the

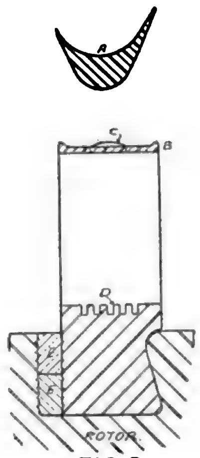


FIG. 8
VICKERS-FERRANTI
NICKEL COATED M.S. BLADE ELECTRICALLY WELDED TO M.S. FOUNDATION RING.
"A" BLADE SECTION. "B" SHROUDING. "C" RIVETING.
"D" BLADE PROJECTIONS AND BASE SERRATIONS.
"E" SIDE CAULKING STRIPS.

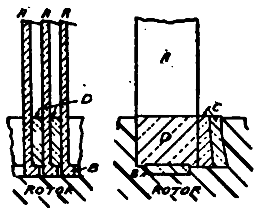


FIG. 9.
WESTINGHOUSE BLADE.
"A" BLADE. "B" BLADE UPSET
"C" SIDE LOCKING RINGS.
"D" DISTANCE PIECE BETWEEN BLADES.

bottom of the blade is cut to form narrow projections. The blades are fed singly into the holder of the welding machine, an electric current passed through the blade, heating the reduced section to a welding heat and the blade is pressed hard home, uniting it to the rotor or foundation ring. Some time ago the writer had the pleasure of making some pulling tests on this type of fastening and found the weld to be as strong as the blade itself. If reblading had not to be taken into consideration and the blades were welded directly to the rotor, we would have a fastening equal in strength to the blade. As mentioned above, for marine work it is highly desirable to

weld the blades on to segments and secure these segments to the rotor. This constitutes the weakest link in the chain. In the list of tests further on in this paper 100 per cent. efficiency was assumed at the weld, and a piece of mild steel of a thickness equal to one blade and a distance was secured to the holder by means of a brass side caulking piece, and pulled to determine the holding power.

The Westinghouse Machine Company, of Pittsburgh, are the patentees of the fastening shown in Fig. 9.

As in the Parsons method, the distance piece forms the anchor. The upset at the bottom of the blade provides a reliable connection between the blade and the distance pieces. The side pieces provide side grip for the blades and secure the distance pieces. They are made up of tapered wedges which facilitate assemblage and repairs.

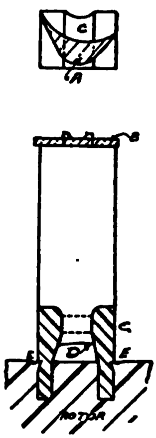


FIG. 10
FORE RIVER FASTENING
"A" SECTION THROUGH BLADE. "B" SHROUDING.
"C" CHANNEL BASE. "D" RIVETING. "E" ROTOR CAULKING.

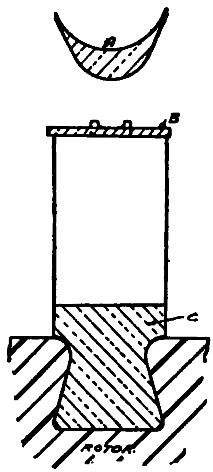


FIG. 11.
PLAIN DOVE-TAIL FASTENING.
"A" BLADE SECTION. "B" SHROUDING
"C" DISTANCE PIECES BETWEEN BLADES.

Fig. 10 shows the Fore River arrangement. It consists of lengths of steel channel-punched at intervals to receive the blades, which are riveted in. The segments of blading are then bent to the curvature of the rotor and inserted in grooves

in the rotor. Caulking of the sides of the grooves holds the blading in position. It is a particularly clever scheme, but in its present form the part of the channel which receives the blades is not thick enough, in the writer's opinion, to provide the length of steadyment necessary to keep down the blade deflection within safe limits. With such a short steadyment the demands made upon the riveting are severe.

The plain dovetail fastening shown in Fig. 11 is now being used by the Fore River Company.

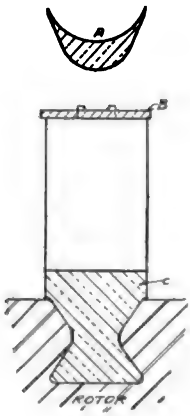


FIG. 12
"A. E. G." FASTENING.
A BLADE SECTION. B. SHROUDING
C DISTANCE PIECES BETWEEN BLADES

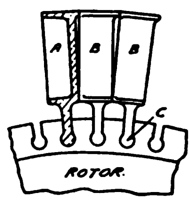


FIG 13
DE-LAVAL FASTENING.
"A" SECTION THROUGH BLADE.
"B" OUTSIDE VIEW OF BLADE. "C" BULB.

Fig. 12 is favored by The Newport News Company. The idea of the double dovetail is to avoid having an abrupt change of blade section at the top of the rotor groove.

The DeLaval fastening is an interesting and well tried-out method. The blades are of nickel bronze and are drop forged with shrouding and roots in one piece. The roots are bulb shaped and are slightly tapered transversely. The rim of the rotor disc is slotted at intervals to receive the blades, which are driven in as shown in Fig. 13.

The blades are not riveted to the shrouding nor are they riveted to the disc. The driving fit used for holding the bulb shanks makes the replacement of blades a simple matter, with-

out danger of distorting the wheel or throwing it out of balance. This result depends upon careful and exact fitting.

The fastening shown in Fig. 15, has not, so far as I know, been adopted on any commercial unit, but its ingenuity is worthy of mention. A dovetailed groove is cut in the rotor with its base forming a wedge and the pressing home of the blades and distance pieces automatically spreads them into the dovetail, filling up the groove. Dr. Jude, of Manchester, England, claims the originality of this scheme.

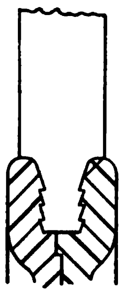


FIG. 14
EARLY DE-LAVAL FASTENING

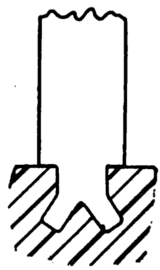


FIG. 15
FASTENING WITH WEDGE BASE

Much credit is due to Mr. L. D. Lovekin, of the New York Shipbuilding Company, for the excellent blade fastening he has recently devised, especially in view of the fact that so many designers' minds have been working on this important detail for so many years. Its sound mechanical construction is worthy of the serious consideration of turbine engineers looking for a rigid fastening of great holding power. It is particularly suitable for high-impulse work.

In Mr. Lovekin's method, Fig. 16, the rotor or disc, as the case may be, is grooved with a plain dovetailed groove into which is inserted a tongued locking piece about twelve inches long. Alternately the blades and the distance pieces, whose roots are milled to a dovetail on one side and a recess on the other, are threaded on to this locking piece until the length is made up. Another piece of locking strip is inserted and more blades and distance pieces threaded on and the operation re-

peated until the entire circumference of the groove is filled up with the exception of a piece of locking strip equal in length to two blades and two distance pieces. This length is closed up as shown in Fig. 17 or 18. After assembling one complete row the locking ring is caulked well down to make sure that the blading completely fills the groove. Mr. Lovekin's side-locking piece is so designed that in the process of caulking the roots of the blades and the distance pieces are first brought in contact with the bottom of the groove and then the blades and distance pieces are forced bodily against the side of the rotor groove. It will be noticed that the bottom of the locking piece is undercut, so that, in caulking, a spreading

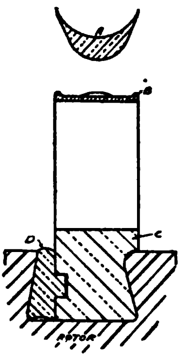


FIG. 16.
LOVEKIN FASTENING
"A" BLADE SECTION. "B" SHROUDING.
"C" DISTANCE PIECES BETWEEN BLADES. "D" LOCKING PIECE.

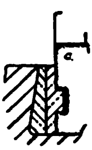


FIG. 17.

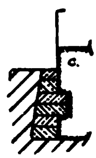


FIG. 18.

CLOSING-UP PIECES FOR LOVEKIN METHOD.

action takes place at the bottom and tightens the lower portion of the blading as well as the upper. The idea has proved very effective. The first three stages of the U. S. torpedo-boat destroyer *Downes'* turbines were bladed in accordance with the Lovekin method, and the finished result presented rows of blading as perfectly in line as could have been obtained had each blade been trued with a square.

The admirable features of the Lovekin method are:

1. Its rigidity;
2. Its great direct pulling strength;

3. The fact that heavy caulking has not to be resorted to to give it its direct pulling strength;

4. That full advantage has been taken of the best known method of assuring rigidity, namely, side caulking;

5. The blading operation is quick, and so simple that the services of experienced bladers are not essential;

6. It is fool-proof;

7. No special recesses are required around the circumference of the rotor grooves for the insertion of the blades—an item of importance in reblading;

8. Extended distance pieces, so valuable in strengthening the blades against bending and providing a good steam line across the stage, can be used;

9. Perfect alignment of the blades in a row is assured;

10. There is no abrupt change of section of blade at the surface of the rotor.

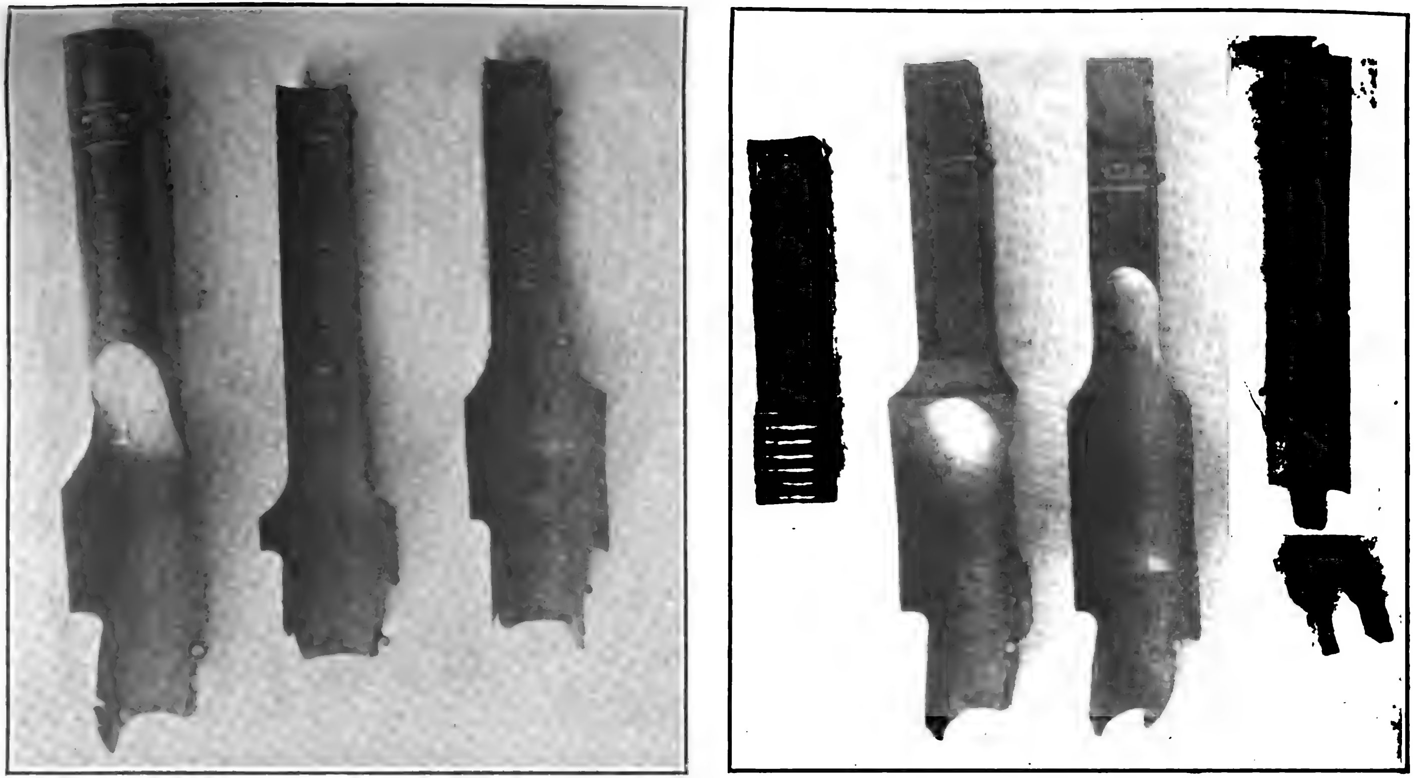
Realizing the many advantages of the Ferranti method, such as eliminating variable circumferential expansion and obtaining an absolute fit of all sectional-base rings by side caulking, "instead of depending on a machine fit which is impossible in manufacturing," caused Mr. Lovekin to consider the Ferranti method as the most advanced or ideal method yet produced. Therefore, he set to work to design a means of holding turbine blades that would have all of the advantages of the Ferranti method without the disadvantage of welding blades to the base rings, which, as is well known, is difficult and costly and beyond the scope of the engineer at sea to repair in case of breakage. The Lovekin method "when made of Monel metal blades with distance pieces combined and having a Monel metal locking ring, which is caulked solidly after all blades have been fitted," enables a result to be obtained that has heretofore been unheard of. The coefficient of expansion is almost the same as that of steel. Then, again, the simplicity of manufacture is such that anyone can blade or reblade a turbine, whether at sea or in the machine shops. This is of paramount importance.

Realizing that it is unnecessary in the lower circumferential speeds such as are used on marine turbines of the direct-drive type (or without the interposition of gearing) to provide the maximum strength of blading, tests were made with the Lovekin method of blading with various modifications. While the same sectional view as shown in Fig. 16 held good for all the test pieces, the actual construction differed. Test No. 7 was made with a brass blade and distance piece combined, using a locking piece of brass. Test No. 12 was made with a brass blade and separate brass distance pieces; but instead of milling out the blade recess the metal was pressed over to engage in its adjacent distance piece. Two other test pieces were made up having a brass blade and separate brass distance pieces, with a ¼-inch diameter wire through the roots, as shown in Nos. 9 and 11. Tests No. 1-A and No. 2 were made with a brass blade and separate brass distance pieces. There are also shown test pieces of the Parsons double-caulked method, No. 8; the Fore River method, No. 6; the fastening as used on the 1st stage of the Argentine battleship *Moreno's* turbines, No. 10; the AEG, Nos. 14 and 15, and the plain dovetail, No. 16. The disadvantage of these three latter methods lies in the great accuracy required in manufacture; while the disadvantage of the Parsons method is its limited application for high centrifugal speeds.

The blade sections used were the latest Fore River Curtis 1.3 inches wide, with the exception of Test No. 6, which was 1 inch wide. It will be noticed that the majority of the test pieces were cut from the bar, the object being to provide a suitable end to accommodate the testing-machine grips.

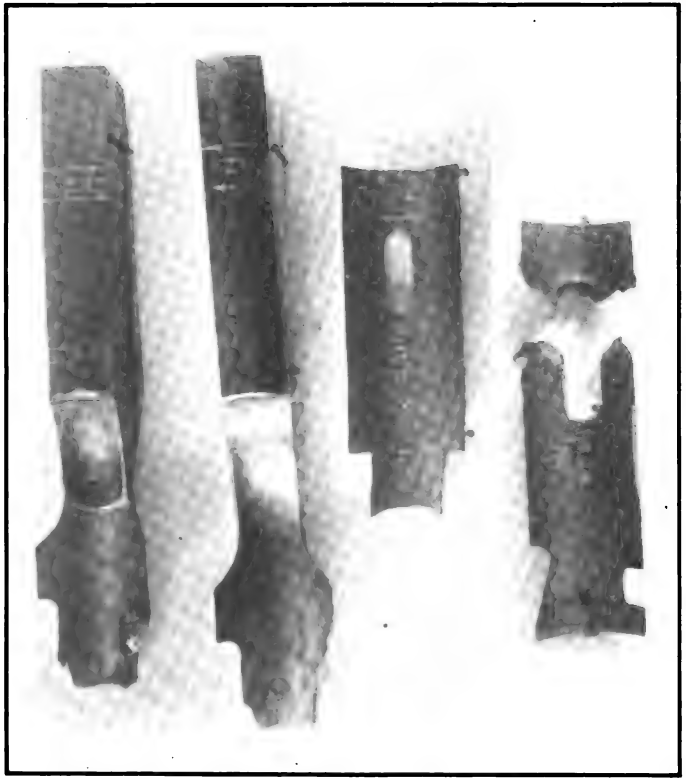
Figs. 1A.T. to 16 T, are photographic reproductions of the test pieces as they appeared after pulling.

In all the Lovekin tests the metal below the recess and the metal at the root of the dovetail was drawn and eventually torn away. The bottoms and sides of the tongues on the locking pieces were locally indented about .01 inch in way of the blades.

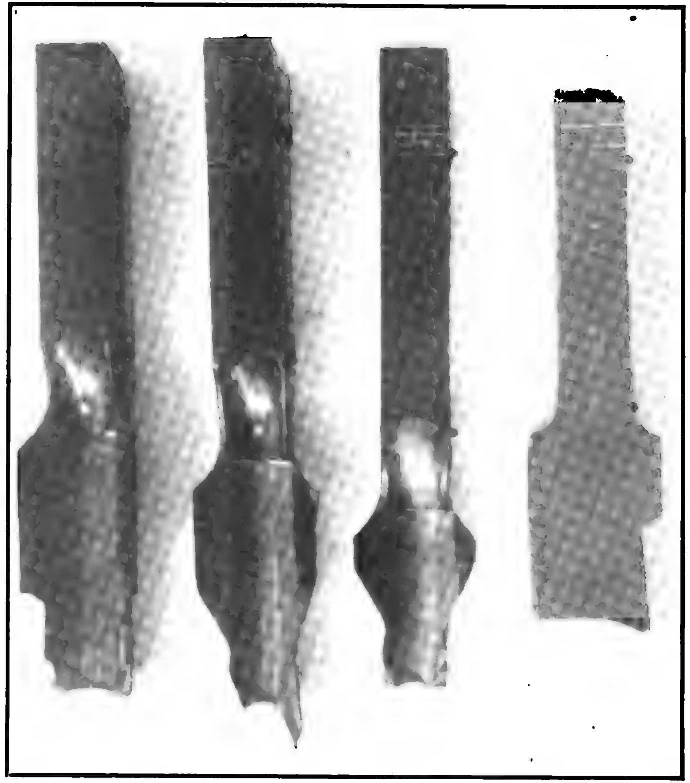


4T 2T 1AT 8T 7T 5T 6T

PHOTOGRAPHIC REPRODUCTIONS OF TEST PIECES AS THEY APPEARED AFTER PULLING.

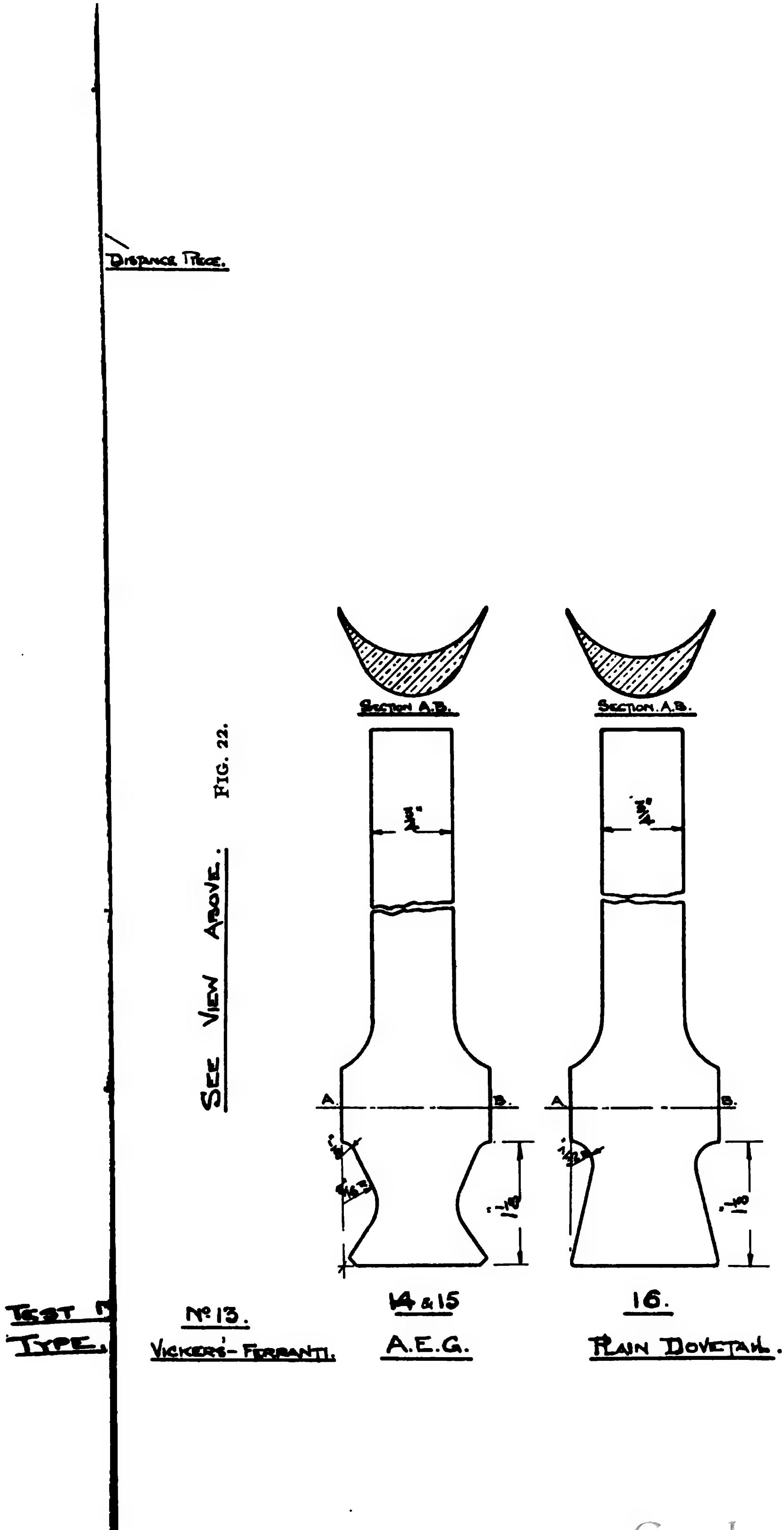


12T 11T 10T 9T



16T 15T 14T 13T

PHOTOGRAPHIC REPRODUCTIONS OF TEST PIECES AS THEY APPEARED AFTER PULLING.



Distance Piece.
Section A.B.
Section A.B.
A.
B.
A.
B.
Test N
Type.
No 13.
Vickers'-Ferranti.
14 & 15
A.E.G.
16.
Plain Dovetail.
See View Above.

Fig. 22.

RESULTS OF TESTS ON VARIOUS METHODS.

Test No.	Piece No.	Description.	Method.	Pounds pull required.		Locking piece.
				To start blade moving.	To pull blade entirely out.	
13	13T	Steel foundation piece with side caulking pieces.......	Vickers-Ferranti.	......	10,700	Brass.
5	5T	Monel blade, solid, with distance piece..................	Lovekin.....	16,900	25,200	Monel.
7	7T	Brass blade, solid, with distance piece......................	Lovekin.....	14,000	17,500	Brass.
4	4T	Monel blade with separate Monel distance pieces.......	Lovekin.....	9,700	12,300	Monel.
11	11T	Brass blade with separate D.P.'s with $\frac{1}{4}$-inch diameter wire through root.	Lovekin.....	8,000	10,500	Brass.
9	9T	Brass blade with separate D.P.'s with $\frac{1}{4}$-inch diameter wire through root.	Lovekin.....	Broke at reduced	section with 9,800	Brass.
12	12T	Brass blade with punched tongue, with separate brass distance pieces.	Lovekin.....	8,500	10,100	Brass.
2	2T	Brass blade with separate brass distance pieces...........	Lovekin.....	5,000	7,000	Brass.
1A	1AT	Brass blade with separate brass distance pieces...........	Lovekin.....	5,000	6,800	Brass.
6	6T	1-inch brass blade riveted to M. S. channel base.........	Fore River.	1,500	2,400	...
8	8T	Brass blade with double caulked distance pieces.........	Parsons.....	3,400	A continued diminishing load.	...
10	10T	Brass blade with separate distance pieces.........	*Moreno*......	3,700	4,700	...
14	14T	Brass blade with separate distance pieces.........	A. E. G......	1,500	9,400	...
15	15T	Brass blade with separate distance pieces..................	A. E. G......	2,500	8,700	...
16	16T	Brass blade with separate distance pieces..................	Plain dovetail.	2,400	8,700	...

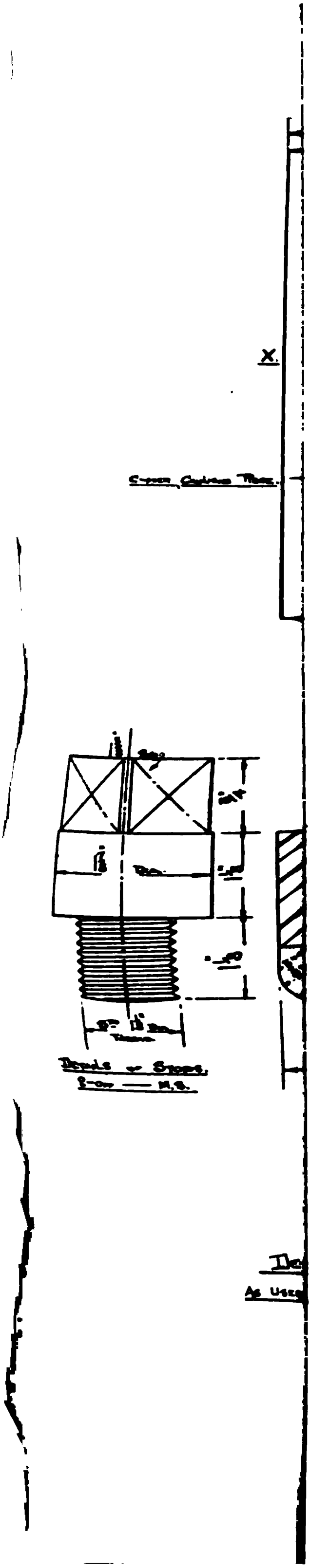
All these test pieces were made up at the New York Shipbuilding Co. and pulled to destruction at Wm. Sellers & Co.'s works on their 100,000-pound Emery testing machine, in the presence of Mr. W. R. Church, of Wm. Sellers, and Messrs. F. F. Kauffman and Jas. A. Capstaff, of New York Shibuilding Company.

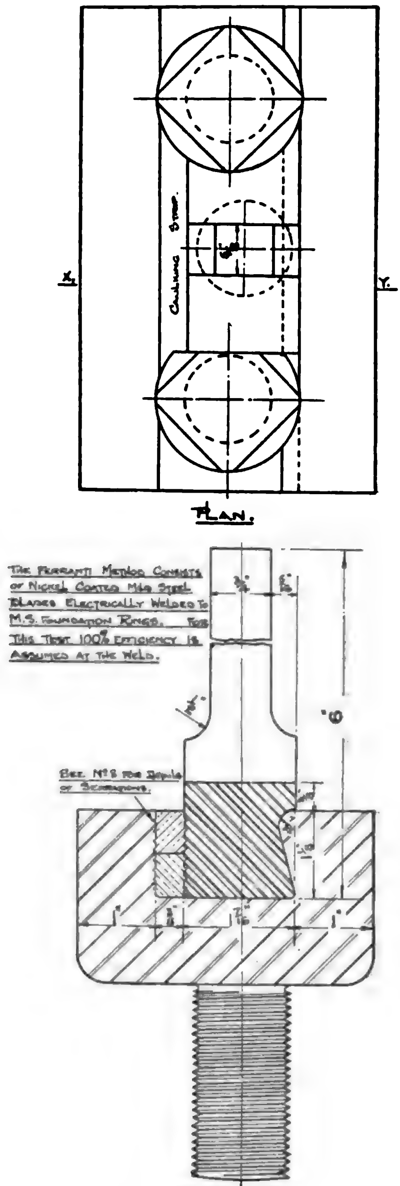
In the Parsons' test the metal of the distance pieces, which was caulked into the vee serrations of the blades and holder grooves, was sheered off.

In the Vickers-Ferranti test the root of the dovetail was drawn, and sheering of the caulked metal took place as in the Parsons test.

The riveted root of the Fore River fastening was drawn parallel.

The other specimens pulled out after drawing and sheering, as will be clearly seen in the photograph. The test-piece holder was of mild steel ,in accordance with Figs. 19 to 22. The projecting piece of the holder was screwed into the weighing head and the blade shank gripped in the jaws of the straining head of the testing machine. The load was applied by the hydraulic straining cylinder, which received its power from a suitable fluid-pressure supply. The essential peculiarity of the "Emery" testing machine is the method by which the stress produced upon the piece being tested is conveyed to the scale and accurately weighed by mechanism that is entirely frictionless, and hence responds equally to the same increment of load regardless of the amount of strain upon the specimen. This result is accomplished by receiving the load upon a flexible diaphragm in a closed cylinder on a hydraulic support. One of these is placed in the scale and a proportionately large one in the machine itself. The pressure received on that in the machine forces liquid to the scale, through a tube, which, acting upon the diaphragm of much smaller area, produces sufficient motion to move the beam of the scale. The general scheme is indicated in Figs. 23 and 24, which shows merely the relation of parts, no attention being paid to proportions. In Fig. 24 "A" is the hydraulic support in the weighing head of the machine, which is connected through a metallic tube to the reducing chamber "B" in the scale. The pressure of the fluid upon the diaphragm in the reducing chamber is transmitted to the large lever "C," and thence through levers and connections to the indicator "F." Suspended from the lever "E" at suitable intervals are poise frames "N," which work

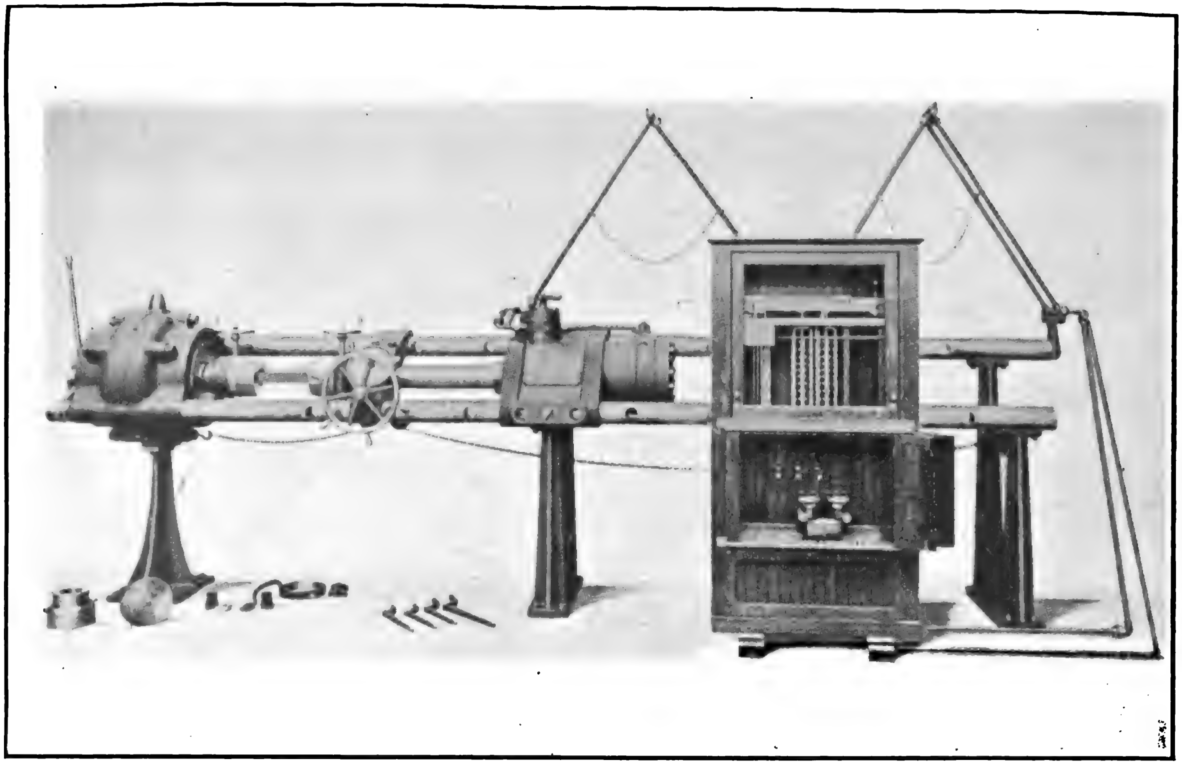




Section on X.Y.

Details of Blade Holder,
As Used for Test No 13.
Vickers – Ferranti.

FIG. 22.



Capable of testing specimens 12 feet long in compression, and 8 feet, 4 inches long in tension. Maximum stroke of piston in straining head, 24 inches. Side bars arranged on incline for transverse testing, also facilitating access to the specimens. A hand wheel is provided for quickly moving the piston for entering and removing the specimens. Hydraulic scale shown in position. Tension holders and specimen in place. The compression platforms are shown lying upon the floor with other fixtures.

FIG. 23.—100,000-POUND TESTING MACHINE.

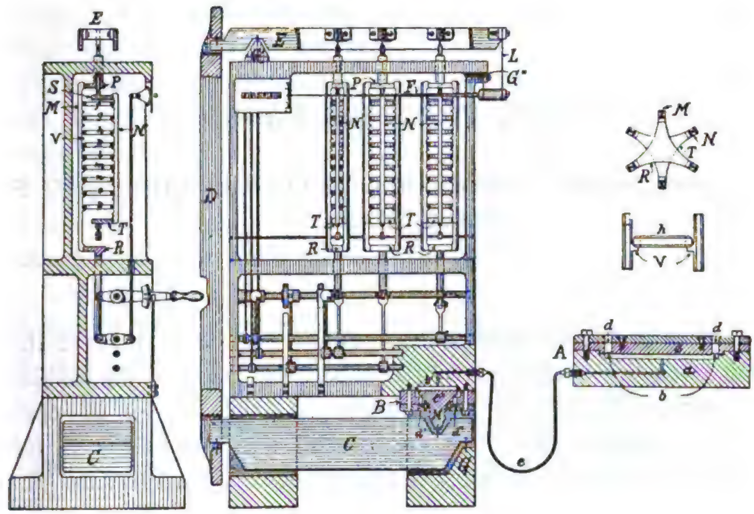


FIG. 24.—EMERY TESTING MACHINE WEIGHING MECHANISM.

in conjunction with the weight frames " M," in such a manner as to permit the weights to be applied in succession by the movement of the hand levers, until an exact balance is obtained. This load applied to the specimen may be read at any time from the numerals which appear in a convenient place on the scale. In place of knife edges, thin steel fulcrum plates are used, which deflect slightly with the vibration of the levers, without measurable friction. Each scale is calibrated with great accuracy, each weight being tested separately in position.

COAL FOR THE NAVY.*

By Lieutenant Commander J. O. Richardson, U. S. Navy, Member.

Owing to the isolated position of the United States and the long stretches between its littoral and the ports of its insular possessions, the Navy Department, in the design of coal-burning vessels, has been confronted with problems which exist in less degree in most of the other principal navies. Its isolation and the comparatively few and widely separated coaling stations made it necessary to devote especial attention to affording the ships the greatest possible coal-carrying capacity. To enable the necessary weight of coal to be carried, it has been necessary to limit to the utmost the weight of armor and of machinery, thereby sacrificing to some extent the vital qualities of protection and speed. The necessity of limiting the weight of, and space occupied by, the boilers has forced upon the Department the installation of boilers of light weight and compact design with limited combustion space.

In order to insure that the coal-burning vessels of the Navy will have the maximum steaming radius, it is necessary that they be supplied with coal that is suitable for use in the boilers on board these vessels and is the best coal obtainable for that purpose regardless of cost.

Section 3728 of the Revised Statutes of the United States provides that, "In purchasing fuel for the Navy, or for naval stations and yards, the Secretary of the Navy shall have power to discriminate and purchase, in such manner as he may deem proper, that kind of fuel which is best adapted to the purpose for which it is to be used."

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* This article is largely compiled from the records of the Bureau of Steam Engineering and from bulletins issued by the Bureau of Mines.

The Navy Department has made it the practice to supply naval vessels with the best semi-bituminous coal obtainable. Strictly speaking, the coal is not purchased under specifications, but the coal delivered must meet the following requirements as to quality:

"Coal to be the best quality, Navy standard steaming, or equal, semi-bituminous, run of mine, with at least 40 per cent. lump, suitable and acceptable for the uses of the naval service.

"Coal must be dry and practically free from slate, dirt, sulphur and other impurities, subject to the usual inspection and test, and must weigh 2,240 pounds per ton, the weight to be determined in a manner satisfactory to the Government."

Navy standard steaming coal is coal supplied from mines on the Navy acceptable list, the delivery fulfilling the requirements as to moisture, percentage of lump and cleanness.

The Navy acceptable list is a list of mines that have supplied coal which has proved satisfactory for use on board naval vessels and whose output is controlled by reliable suppliers that may be depended upon to deliver the required amount of satisfactory coal under all conditions of the coal and labor market.

The question as to the reliability of the supplier is one of vital importance because it is necessary that the Navy be assured of an adequate supply of coal on short notice, and in actual practice it has been found that the large suppliers whose mines are on the acceptable list have supplied coal to the Navy to meet the full requirements when there was a large demand for coal at a price above the Navy contract price.

Mines are liable to suspension or elimination from this list if the deliveries of coal are the subject of complaint by the ships of the fleet or if the supplier fails to fulfill the conditions of his contract.

Each bidder on the naval coal contracts is required to supply the following information regarding the coal he proposes to furnish:

(*a*) Commercial name of coal.

(*b*) Name of mine or mines.

(*c*) Exact location of mine or mines.

(*d*) Name or other designation of the coal bed or beds.

(*e*) British thermal units per pound of "dry coal."

(*f*) Percentage of ash in "dry coal."

(*g*) Percentage of sulphur in "dry coal."

(*h*) Percentage of volatile matter in "dry coal."

(*i*) Moisture in coal as received.

(*j*) Whether or not bidder has absolute control over all of the coal from the mines upon which bid is based.

When the coal which the bidder proposes to furnish does not come from a mine on the Navy acceptable list, the bid will not be considered if the coal has an analysis indicating quality lower than the following:

(*a*) Moisture in "delivered coal," . . 3 per cent.

(*b*) Ash in "dry coal," . . . . 7 per cent.

(*c*) Volatile matter in "dry coal," . . 21 per cent.

(*d*) Sulphur in "dry coal," . . . 1¼ per cent.

(*e*) British thermal units per pound of "dry coal," 14,500,

If the analysis of the coal indicates quality equal to or better than this, the bidder may be awarded a contract for a small amount of coal, the contract being subject to cancellation if the coal proves unsatisfactory for use on board ship, or the amount called for being subject to increase if the coal proves satisfactory, in which case the mines supplying the coal would be placed on the Navy acceptable list and the coal would be classed as Navy standard steaming coal.

The following is the usual and most satisfactory method of determining the suitability of coal for naval use and the eligibility of the mine for the Navy acceptable list.

When the analysis of the coal indicates an acceptable quality the supplier is offered the opportunity of having the coal tested at the Naval Engineering Experiment Station at Annapolis, Maryland, the conditions being that the supplier deliver at the Experiment Station about twenty-five tons of coal, free of cost to the Government, and agree to defray the cost of the test, making a deposit of about three hundred and fifty dollars for this purpose. To insure that the coal tested

represents commercial deliveries of run-of-mine coal from the mine specified the sample is selected by a naval coal inspector or other Government representative.

If this test is satisfactory one cargo of coal is purchased from the supplier for use on board naval vessels and, if the ships using the coal report that it is satisfactory, the mine is placed on the Navy acceptable list, the coal is classed as Navy standard steaming coal, and the supplier has the same opportunity to secure awards of coal contracts as other suppliers of Navy standard steaming coal.

MOISTURE.

The percentage of moisture allowed is small because where coal is purchased on the weight basis the moisture is paid for as coal and is of no value as fuel.

All coal necessarily contains some moisture. It is probable that a small amount of moisture is harmless and possible that a limited amount of moisture is beneficial on account of the water vapor in the furnace facilitating the combustion of the volatile combustibles driven off from the coal. The addition of a small amount of moisture by wetting the coal before it is fired may result in a saving by preventing the fine coal from being blown up the smoke stack and by obtaining better combustion, but this gain may be offset by the increased amount of heat carried away by the water vapor.

Excessive moisture in coal results in a reduction in the capacity of the boiler, caused by a lower rate of combustion, and an increased loss of heat in the smoke-stack gases, due to a slightly higher flue-gas temperature, the amount of heat required to heat the moisture itself and the escape of a larger quantity of free hydrogen.

ASH.

The percentage of ash is limited because ash is an impurity of no value. The loss due to presence of ash in the coal depends as much upon the character as it does upon the quantity of the ash. The most important characteristic of Navy coal

is that it must produce as much power per pound of coal as possible, and, of course, a large percentage of ash reduces the heat value per pound of coal, reduces the steaming radius of the ship using the coal, and throws upon the firemen an excessive amount of work in handling the refuse.

If the ash is fusible it will form clinkers which may cling to the grate bars and reduce or entirely shut off the air from some parts of the grate and, in breaking the clinker from the grate, it may become mixed with coal and cause a large loss. Even if the ash is not fusible it covers the grate and increases the resistance of the fuel bed to the passage of air through it and causes a loss in cleaning the fires.

No reason can be definitely assigned as to why some coal forms clinkers in burning and others do not, even when the percentage of ash is greater. For example, George's Creek coal as a rule contains a higher percentage of ash than Pocahontas does, yet it forms almost no clinker, whereas Pocahontas forms a heavy clinker that is easily removed from the furnace.

VOLATILE MATTER.

The percentage of volatile matter is one of the most important characteristics of coal, because high volatile coal is not adapted for use in Naval boilers, and this fact is one that is hard to impress on many dealers, because a high volatile semi-bituminous coal may be very high in heat value and, judged purely on a heat unit basis, it would be ranked very high as fuel for any purpose.

The percentage of volatile matter is limited because the combustion space is restricted, the coal must be hand fired, and forced draft is necessary to produce the requisite rate of combustion for the development of full power.

Coals high in volatile matter can be burned efficiently where natural draft, and mechanical strokers giving a slow rate of heating to the coal, are used in connection with boilers having ample combustion spaces; but to produce a fairly good efficiency with high volatile coal with hand firing re-

quires far greater care and skill than is required with low volatile coal.

In hand firing the coal is heated through a range of about 2,400 degrees F. in two or three minutes. Experiments have shown that, with any given coal, the amount and quality of the combustible driven off the coal by heating depend largely upon the rate of heating, and that when the rate of heating the coal is rapid, as in hand firing, the total volatile matter driven off is not only high in quantity but contains much tar vapor and, of course, this condition is greatly aggravated when the coal is high in volatile matter.

The tar vapors and other heavy carbon-hydrogen compounds which are the products of distillation of coal, when heated rapidly, burn slowly and, in order to burn them completely, they must be kept a comparatively long time within the furnace. When burning coal in a Babcock & Wilcox boiler, as installed on the *Delaware*, for example, at the rate of 20 pounds of coal per square foot of grate surface per hour, with the fuel bed one foot thick, the time that each cubic foot of gas stays in the furnace is about .25 seconds, so that a large volume of smoke is unavoidably emitted, and, when the coal is high in volatile, this smoke is dense and black and contains a large amount of tar vapor and other carbon-hydrogen compounds which escape unburned with a resulting loss and the accompanying military disadvantage of a heavy cloud of smoke that is visible many miles.

When forced draft is used the rate of combustion is increased and the volume of volatile matter driven off greatly increased, so that the time available for the combustion of the products of distillation is greatly reduced and the speed of the gases through the tube space so increased that they reach the uptakes before their temperature is reduced below the point of ignition, so that combustion takes place in the uptakes and the smoke stack with consequent overheating and greatly accelerated deterioration of these parts.

Another disadvantage of high volatile coal is its liability to give off inflammable gases while stored in bunkers; and

while a high percentage of a volatile matter may not be a cause of spontaneous combustion of coal, it is a fact that nearly all cases of spontaneous combustion of coal in the Navy have occurred in coal that contained a fairly high percentage of volatile matter.

SULPHUR.

The amount of sulphur is limited because it is an undesirable element in coal and, as it usually occurs in combination as iron pyrites, it is worthless as fuel and, even in the free state, it has a heating value of only 4,000 B.t.u. per pound.

Sulphur has a decided effect upon the clinkering quality of the ash and, as a general rule, the tendency of a coal to clinker varies directly as the sulphur content and inversely as the ash content.

When coal containing a high percentage of sulphur forms clinkers that adhere to the grate, the sulphur attacks the grate bars and causes them to burn away and deteriorate rapidly.

The clinker may sometimes be prevented from running between the grate bars by blowing steam under the grate, this result being due to the fact that the steam is decomposed by the heat, and the cooling process of decomposition reduces the temperature of the clinker below its melting point.

The presence of sulphur in coal may be a contributary cause of spontaneous combustion, but opinion is fairly well divided as to whether or not sulphur is a cause of spontaneous combustion; however, inasmuch as it may be a cause, this point is mentioned as one reason for limiting the percentage of sulphur allowed.

HEAT VALUE.

The heating value of the coal is the most important quality because, in the purchase of coal, one is buying useful heat units, and in coal for the Navy it is necessary that the coal be high in heat units so fixed in the coal as to become available and productive when the coal is burned in the furnace of

a naval boiler because the vessels must be able to steam the greatest possible distance with the weight of fuel carried, and must be able to attain their maximum speed when it is required.

OXYGEN.

In the description of coal that might prove acceptable to the Navy no mention is made of oxygen, but a limit is placed upon the percentage of this element that is permissible by the specifications as to ash and heating value. The anticalorific value of oxygen in coal is practically equivalent to the anticalorific value of the same percentage of ash, so that even if the ash is low the heating value will be low also if the oxygen is high.

In addition to oxygen having the same anticalorific value as ash, its presence in coal is further objectionable because the oxygen content is an index of the avidity with which coal absorbs oxygen, high-oxygen coal absorbing oxygen more rapidly than low-oxygen coal. Spontaneous combustion of coal is due to the oxidation of coal in places where the heat formed cannot be dissipated, so that a coal which has a great avidity for oxygen is a coal that is liable to suffer from spontaneous combustion.

THE PURCHASE OF COAL ON A B.T.U. BASIS.

The Navy Department has been frequently urged to purchase coal under strict specifications that provide that the purchase price shall be a certain figure for coal that shows a heating value of a certain number of British thermal units per pound; that coal which falls below or exceeds in calorific value this number shall be purchased at a price less or greater than the base price by an amount depending upon the number of B.t.u. below or above the standard.

Theoretically, this is an ideal way of purchasing coal, but, practically, it is very unsatisfactory, because it frequently happens that coal which, in a laboratory test, would be rated very high on a B.t.u. basis, gives very poor results when

burned in the furnace of marine boilers, and conversely. In fact, one of the very best steaming coals on the market, and one which is largely used by sea-going steamers, would be barred from competition because it is lower in B.t.u.'s than other first-class coals and would, therefore, be forced to accept a lower price than such coals or abandon the Navy trade, which it probably would. This coal, George's Creek, has long been recognized by marine engineers as one of the most satisfactory on the market.

"The method of purchase adopted by the Navy is the result of experience in the actual burning of coal in marine boilers over a period of seventy years, and was not adopted without careful consideration of all the questions involved."

This method of purchasing coal would no doubt cause many disputes between the suppliers of coal and the Navy Department, and, in the end, would be unsatisfactory, because the Navy is not buying so many heat units but is buying high-grade coal that by actual use has been shown to be the best coal obtainable for use in Naval boilers, and the suitability of the coal for the duty required should be the controlling factor in the purchase of coal for use on board naval vessels.

WEST COAL COALS.

Practically all the coal for the Navy comes from the eastern fields which supply Pocahontas, New River, George's Creek and Eureka coal, but the Navy Department, for military and economic reasons, has endeavored to find a coal mined near the Pacific Coast that would be acceptable for the uses of the naval service.

Coal from British Columbia, Washington, Colorado and Utah, has been tested, but no coal has been found that is as satisfactory as the Navy standard steaming coals, and the investigations of the Bureau of Mines and of the Geological Survey fail to show any evidence of the existence of a western coal that would be satisfactory for naval purposes.

All western bituminous coals have a relatively high per-

centage of volatile matter. Many of the best have a large percentage of ash, and most of those tested on board naval vessels contained ash that formed a heavy clinker that adhered to the grate bars and bridge wall.

It has been said that Western coal would give better results if a different form of grate bar were used, but it is probable that the improvement, if any, resulting from the use of these bars would be slight; and there are serious objections to their use. Naval vessels may be required to proceed on short notice to places where Western coals are not obtainable. It would, therefore, be necessary for every vessel to carry two complete sets of grate bars—one for burning Western coals and the other for burning Eastern and Welsh coals. A set of the usual type of grate bars for the *Colorado* weighs, including ten per cent. of spares, about fifty tons. There would be no place to store these bars except in coal bunkers, where they would take the place of coal, and would still further reduce the coal-carrying capacity. The set of special bars would cost, with the necessary spares, about $3,000 for each ship of the *Colorado* class.

It is claimed that fairly good results would be obtained with Western coals if the firemen were especially trained to burn this coal. This is probably true, but it is not believed that it is possible, even with special grate bars, and with firemen skilled in using Western coal, to burn the best of these coals with results approaching those with Eastern coal. Besides, there are decided objections to training men to burn this coal to the exclusion of Eastern coal. In order to teach the men the most efficient method of burning the high volatile Western coals it would be necessary for them to forget what they now know about the use of Navy standard steaming coal, and it would result in reduced naval preparedness, because in time of war the best obtainable coal must be used and the men must be qualified to produce the maximum efficiency with this coal.

The existence of the Navy is justified only on the supposition that it will be kept at all times in the maximum state of

readiness for war. Wars often come with little, if any advance notice, and, if the Navy is allowed to fall into a state of unpreparedness in any detail with the expectation of perfecting that detail when war is threatened, it is inviting disaster. One of the most important of these details is the ability of the fleets to steam the greatest distance with the amount of coal they can carry, and to develop the maximum speed when required. This ability depends in a large degree upon the skill of the fireroom force, and this skill can be assured only by careful instruction and training. It is essential to the country's interest that the firemen on board its men-of-war should be kept as well trained as the gun crews. They are at present well trained. To make them unlearn what they now know, learn to handle a kind of coal which is to be used only in time of peace, and then depend upon their relearning, in time of threatened hostilities, to burn to best advantage the kind of coal that must be used in time of war, would be analogous to requiring the gun crews to use brown powder at target practice and depending upon them to learn, at the outbreak of war, to use smokeless powder, the only suitable kind in battle.

ALABAMA COAL.

There are coals in Alabama that are high in heat units and low in ash and sulphur, but all of them are relatively high in volatile matter. Several tests have been made on board ship with Alabama coal and the results were not unsatisfactory when burned in Scotch boilers under natural draft, but it is believed that the percentage of volatile matter is so great as to preclude the possibility of coal from any of the known mines proving satisfactory as Navy standard steaming coal.

ARKANSAS COAL.

There is coal in Arkansas whose analysis is such as to indicate that it would be satisfactory for naval use, and an evaporative test of this coal is to be made at the Naval En-

gineering Experiment Station in the near future. If this test should prove satisfactory it is possible that an additional source of supply for coal for naval use may be developed.

ALASKA COAL.

Under the direction of the Navy Department, tests have been made on board the U. S. S. *Maryland* of samples of coal from two fields in Alaska.

The tests of the Bering River coal in 1913 showed conclusively that coal from the mine (location) selected for test was entirely unsuitable for naval use.

The test of the Matanuska coal in 1914 showed that the coal represented by this sample is suitable in every respect for use in the naval service.

There are many kinds of coal in the Bering River field, and it is very probable that when this field is developed a coal will be produced that will be suitable for naval use. Coal from one entry in this field shows the following results on analysis:

Moisture (as received), . . .	2.68 per cent.
Volatile matter (dry coal), . .	16.70 per cent.
Fixed carbon (dry coal), . .	78.40 per cent.
Ash (dry coal), ...	4.90 per cent.
B.t.u. per pound (dry coal), . . .	14,962.

If there is coal in Alaska that is suitable for use in the naval service, this source of supply is protected, because the "Act to provide for the leasing of coal lands in Alaska, and for other purposes," approved October 20, 1914, carries the following provision:

"SEC. 2. That the President of the United States shall designate and reserve from use, location, sale, lease or disposition not exceeding five thousand one hundred and twenty acres of coal-bearing lands in the Bering River field and not exceeding seven thousand six hundred and eighty acres of coal-bearing

land in the Matanuska field, and not to exceed one-half of the other coal lands in Alaska: Provided, that the coal deposits in such reserved areas may be mined under the direction of the President when, in his opinion, the mining of such coal in such reserved areas, under the direction of the President, becomes necessary, by reason of an insufficient supply of coal at a reasonable price for the requirements of Government works, construction and operation of Government railroads, for the Navy, for national protection, or for relief from monopoly or oppressive conditions."

REPORT OF THE U. S. NAVAL RADIOTELEGRAPHIC LABORATORY.

L. W. AUSTIN, HEAD OF THE LABORATORY.

An account of the work of the laboratory from its beginning in 1908 to 1912 was presented to the Chief of the Bureau of Steam Engineering in 1912 and published in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXIV, p. 122, 1912. The present report deals with the work which has been done since that time.

Most of the experimental work not involving large pieces of apparatus or high power has been done at the laboratory at the Bureau of Standards, and two chief electricians have been assigned as assistants in this laboratory and a large number of others, both chief electricians and civilian radio aids, have assisted in the experiments involving sending and receiving at a distance. At present there is but one laboratory assistant as the other has been employed for the last four months in connection with the operation of the Tuckerton Radio Station.

The work of the laboratory may be divided into three parts:

1. *Routine testing* of small apparatus offered to the Department for use in radio telegraphy. The following is a list of these tests:

Complete receivers	23
Condensers	32
Inductances	12
Detectors	16
Telephones	19
Wavemeters	8
Amplifones	11
Detector testers	4
Ammeters	6
Audibility boxes	15
Buzzers	1

2. *Instruction* of a limited number of naval electricians in the practice of high-frequency measurements. For this work a typewritten course of instruction has been prepared which it is expected will soon be revised and printed for the general use of the service. The number of men who have taken this training course up to the present is 67. This part of the work is at present discontinued on account of the lack of assistants.

3. *Scientific work,* involving the general experimental study of high-frequency phenomena. Since the last report the main scientific work of the laboratory has been along the following lines:

1. Energy losses in condensers (equivalent condenser resistance.)
2. The high-frequency resistance of inductances.
3. Antenna resistance.
4. Experiments in the efficiency of different antenna types.
5. Quantitative experiments on radiotelegraphic transmission.
6. The transmission efficiency of continuous and damped waves.
7. A study of the ultraudion circuits.
8. Prevention of atmospheric disturbances.

Articles on all the subjects either have been or are about to be published as noted in the list of publications at the close of this report. A brief resumé of these articles is given below.

1.—ENERGY LOSSES IN CONDENSERS.

This work was begun several years ago and an account of the general methods employed, was given in the report already cited. The most important results obtained since the last report have to do with the change of effective condenser resistance with the frequency. In this work the method employed was that of measuring a current in a high-frequency circuit, first with the unknown condenser in circuit and then with a variable air condenser of the same capacity. Fine wire resistance was then placed in series with the air condenser

until the current in the circuit was the same as that observed when the unknown condenser was in circuit. The series resistance then represented the equivalent resistance due to the dielectric losses in the condenser being tested. As the dielectric losses are in most cases independent of the voltage and as it was desired to keep free from brushing losses, the condenser circuit was excited by a buzzer-driven wavemeter.

TABLE I.*

Relation of Resistance and Wave Length in Glass Condenser.

Wave Length, meters.	Resistance, ohms.
385	0.7
650	1.3
910	2.0
1,325	3.0
1,905	4.8
2,360	5.4
3,100	7.5

Table I gives the equivalent resistance of a copper-coated glass-plate condenser of 0.00196 mf. capacity. If the results be plotted it will be seen that the resistance increases directly as the wave length. See Fig. 1.

2.—THE HIGH-FREQUENCY RESISTANCE OF INDUCTANCES.

While the high-frequency resistance of non-inductive resistances can be easily determined by substitution methods, the determination of high-frequency resistance of inductances offers great difficulties.

The principle of the method used is briefly the following: Two equal inductance coils are placed in identical oil calorimeters, one coil is heated by a high-frequency current and the

* Proc. Inst. of Radio Engineers, I, p. 35, 1913.

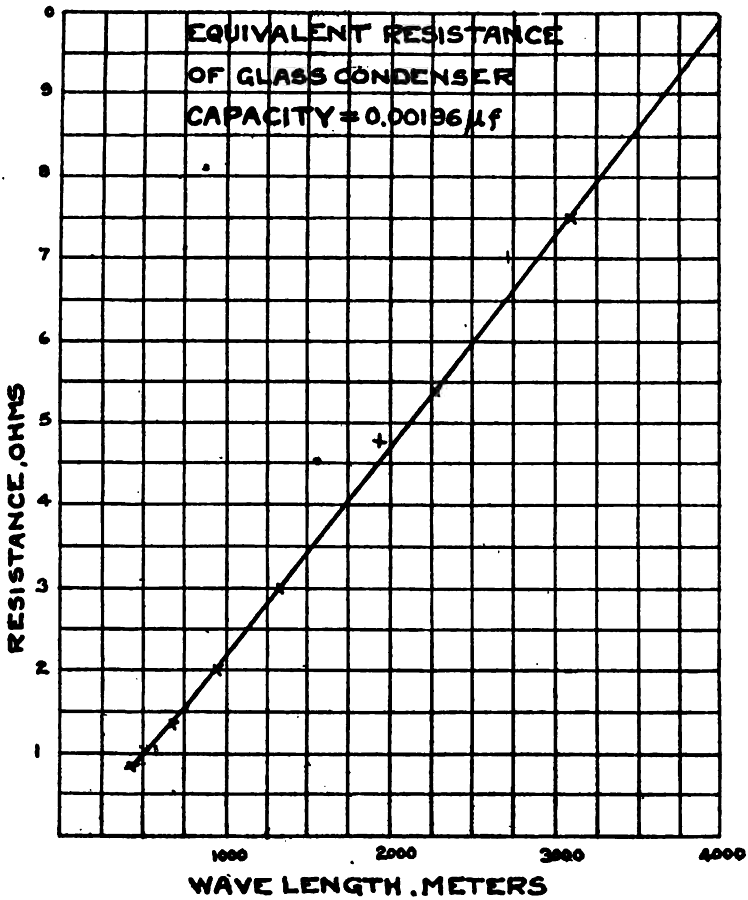


FIG. 1.

other by direct. When the calorimeters are both brought to the same temperature in equilibrium with their surroundings, the heat imparted to each per second must be the same. This heat is proportional to $I^2 R$, and the ratios of the resistances of the two coils at the given frequency and for constant current are inversely proportional to the squares of the currents. To compensate for the slight inequalities in the coils and calorimeters, the direct and high-frequency currents are interchanged and the mean values of the ratios of the current

squares taken. The high-frequency current is produced by a rotary spark gap in an oscillatory circuit coupled loosely to the circuit containing the inductance coil to be measured. The two circuits are brought to resonance at the frequency desired, and the high-frequency current through the inductance, regulated by varying the coupling. The current is read on a non-shunted hot-wire ammeter which has been accurately calibrated for high frequencies. The direct current for the other coil is supplied by a storage battery and the final regulation for equilibrium is made in this circuit. Equality of temperature between the two calorimeters is determined by a differential constant and copper thermoelement. The calorimeters are heated to from 20 degrees to 30 degrees above the temperature of the room. The uncertainty of the individual high-frequency current readings is approximately one part in fifty. The mean of thirty or more readings is taken as a basis for each calculation. The calorimeters, provided with motor-driven stirrers, are of glass, 15 cm. high and 10.8 cm. in diameter, and contain sufficient petroleum to cover the coils under experiment. The coils, of 0.40 mm. double silk-covered copper wire, are wound on glass, and the principal constants are given in Table II and their resistances in Table III.

TABLE II.

Diameter of double silk-covered copper wire = 0.04 cm.
Diameter of coils = 8.6 cm.
Turns of wire per centimeter = 18.9.

Coil.	Length. cm.	Turns of wire.	Inductance. m.h.
1	0.60	11.0	0.022
2	1.10	20.5	0.066
3	1.55	29.5	0.115
4	2.65	50.0	0.273
5	3.75	70.5	0.475
6	5.40	102.0	0.775

TABLE III.

Coil 1.		Coil 2.		Coil 3.		Coil 4.		Coil 5.		Coil 6.	
	R		R		R		R		R		R
m.	ohms.	m.	ohms.	m.	ohms.	m.	ohms.	m.	ohms.	m.	ohms.
360	1.45	550	2.06	780	2.87	780	6.60	1,440	7.75	1,500	13.3
550	1.09	780	1.70	970	2.54	970	5.93	1,900	6.55	2,000	11.8
780	0.86	970	1.51	1,150	2.20	1,440	4.65	2,400	6.28	2,500	10.9
970	0.72	1,150	1.40	1,440	2.02	1,900	4.03	2,900	5.93	3,000	10.2
D.C.	0.42	1,440	1.31	1,900	1.81	2,400	3.52	D.C.	2.68	3,500	9.6
...	...	D.C.	0.78	D.C.	1.12	D.C.	1.90	...	...	4,000	9.1
...	...	...	...	...	...	...	...	...	...	D.C.	3.88

After the determination of the resistance of the six pairs of standard coils a roller inductance of the Fessenden type was calibrated by comparison with the standard coils. The method used is as follows: A buzzer-driven wave meter is used to excite an oscillatory circuit containing a sensitive thermoelement and variable air condenser. By means of switches either one of the standard coils or the variable inductance can be inserted in this circuit, the deflections in each case being observed on the galvanometer of the thermoelement. Sufficient fine-wire resistance is then placed in series with the inductance giving the larger deflection and adjusted until its deflection is reduced to that of the other inductance. The resistance of the standard coil for the given frequency being known, the corresponding resistance of the variable inductance at this point is at once determined. When the variable inductance has been calibrated in this way for several points and at various wave lengths, it at once becomes a standard of comparison of resistance for any other inductances within its limits, by a method similar to the above. If the values of the resistances in Table III for any given wave length be plotted against the number of turns, that is, the length of wire in the different coils, it will be seen that the high-frequency resistance of the coils increases more rapidly than the number of turns. By plotting the resistance against the inductance,

however, the values will be found to be nearly in a straight line.

The increase in resistance of the comparatively fine wire in the above coils due to its being coiled up and not straight is only of the order of three to one. In larger wire the difference is much greater. For example, a fifty-turn coil 7.8 cm. long and 15.9 cm. in diameter wound with 1.29 mm. (No. 16) wire and having an inductance of 0.428 m.h., at a wave length of 1,000 meters has a resistance about seven times greater than that of the same wire straight at the same frequency, and twenty times greater than its direct-current resistance. From this it follows that it is difficult to reduce the resistance of coils below certain limiting values. For the wire sizes used in receiving sets this limit is about 16 ohms per millihenry at $\lambda = 1{,}000$ m. both for stranded* and solid wire. At $\lambda = 3{,}000$ m. the limit is, approximately, 9 ohms per millihenry for stranded wire or solid wire of more than 1 mm. in diameter. Sending inductances are slightly better, giving as a limit about 11 ohms at 1,000 m. and 7.5 ohms at 3,000 m.

3.—ANTENNA RESISTANCE.

A brief account of the early work on this subject was given in the first report. Since that time the resistance of a large number of antennae has been measured. Two characteristic antenna curves are given in Figure 2. That of the Bureau of Standards, which is typical of the resistance curves of all stations with poor grounds, shows a marked fall in resistance as the wave length is first increased, followed by a sharp rise with further increase in wave length. The curve of the U. S. S. *Maine* is typical of ships and land stations with thoroughly well-constructed ground systems. The falling part of the curve which is common to all stations represents the decrease in radiated energy as the wave length is increased. The radiated energy is expressed mathematically by

* The superiority of stranded wire does not show itself below $\lambda = 1{,}500$ m.

$$E = 160\pi^2 \left(\frac{h}{\lambda}\right)^2 I_s^2.$$

Where h represents the height to the center of capacity of the antenna, λ the wave length and I_s the current in the vertical portion of the antenna, the expression

$$160\pi^2 \left(\frac{h}{\lambda}\right)^2$$

is frequently spoken of as the radiation resistance, as it takes the same position in the energy equation as that occupied by R in the case of ohmic losses. It is particularly to be noted that the radiated energy at the sending station and the amount of

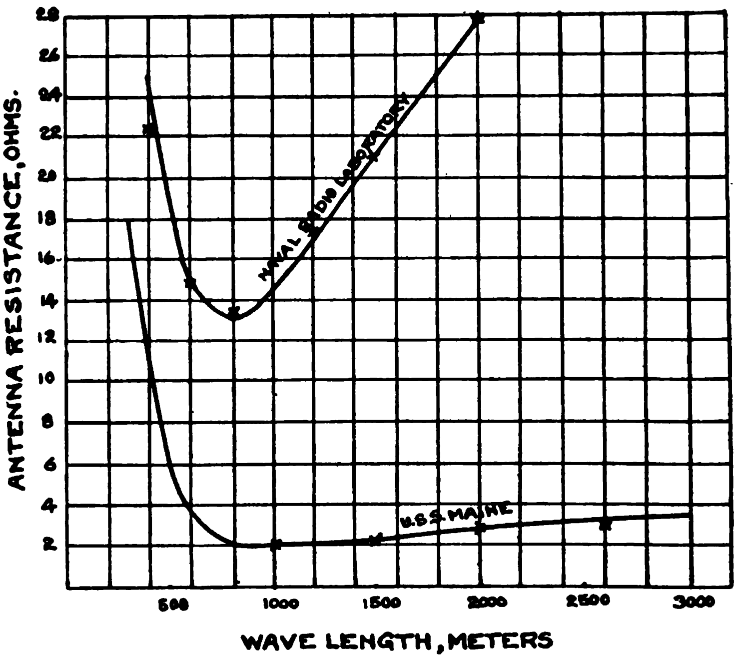


FIG. 2.

electro-motive force impressed on the antenna at the receiving station depend only on the height and shape of the antenna and the antenna current, the wave length remaining constant. The total antenna energy I^2 multiplied by (radiation resistance + ohemic resistance + earth resistance), which is often cited

in connection with wireless installations, has little to do with the actual sending effectiveness of the station, although total energy divided by the energy delivered to the spark gap has a technical interest as indicating the quality of the sending set.

Great difficulty was found at first in explaining the increase of resistance of most land stations with increasing wave length, as shown in the curve of the Bureau of Standards antenna, beyond 1,000 meters. The explanation was finally found in connection with the change of condenser resistance with changing wave length mentioned under condenser losses. Beyond a certain point the antenna resistance rises in proportion to the wave length just as does the condenser resistance in Table I. The connection between the two phenomena is evident when we remember that an antenna must be looked upon as a condenser, the antenna itself being the upper plate and the ground water being the lower. The greater part of the dielectric is, of course, air which is free from dielectric losses, but in most cases above the ground water there is a semi-conducting mass of earth which is a poor dielectric and undoubtedly furnishes the dielectric losses which produce the rise in the antenna resistance curve at the greater wave lengths. If the ground under the antenna be sufficiently well covered with wire this effect nearly disappears.

4.—EXPERIMENTS ON THE EFFICIENCY OF DIFFERENT ANTENNA TYPES.

In these experiments an attempt was made to gain information regarding the efficiency of different antenna types, using ordinary radiating wire grounds, such as are furnished with the portable sets, and various kinds of counterpoises. The antennas were supported on 40-foot portable masts erected on low, wet land near the Arlington station, thus ensuring good ground conditions.

The types of antennas used were: a triangle 85 feet on a side supported by three masts, a "T" 85 feet long, an "L" 85 feet long, an umbrella of six 85-foot wires, and an umbrella

of eight 40-foot wires. Each of the umbrellas was tried with its wires extended at 73 degrees to the vertical, and again at 45 degrees. In all the experiments the wave length was 1,000 m. and a current of approximately 2 amp. was maintained in the different antennas. Receiving measurements were made at the Radio Laboratory by means of thermoelements. Receiving experiments were also made on the antennas under test, the sending station being the Washington Navy Yard. The full data of the experiments are too extensive and involved to give here. The results may, however, be summarized as follows:

The triangle, the "T" and the "L" gave the best absolute results, both in sending and receiving, on account of their greater effective height. The 40-foot wire umbrella showed itself superior to the standard 85-foot type. The relative efficiency of the 40-foot wire umbrella, taking into account its lower effective height, is greater than that of any of the other antennas.

In some of the experiments with the umbrellas the ground net was replaced by a circular counterpoise of from two to six wires running around just outside the ends of the antenna wires, leaving the ground directly below the antenna bare. As the counterpoise was insulated, it was hoped to throw the electric field outward, thus producing a greater effective height. This object was to a certain extent attained in sending, the efficiency of the 40-foot wire antenna at 73 degrees being raised from 52 per cent. to 63 per cent.

An experiment was also made making use of a ground formed by driving pipes into the earth so as to get direct conductive connection with the ground water. The results, however, were found to be extremely poor compared with the net grounds and counterpoises.

In connection with this work it was desired to investigate the effect of metal towers used in supporting antennas. These were simulated electrically by extending wires from the top of the masts to the ground net. In the case of the triangle antenna it was found that the sending efficiency was reduced

20 per cent., and the receiving efficiency 11 per cent. by the presence of these artificial grounded towers.

In the experiments on directive effects it was found that it made no certain difference whether the "L" antenna was used in sending to the Radio Laboratory with its open end toward the Laboratory or away from it. Extending a counterpoise from the base of the "L" antenna 280 feet toward the Radio Laboratory was found to give a slightly improved received current. In the case of the 40-foot umbrella with antenna wires at 45 degrees a directive counterpoise was constructed extending 260 feet toward the Radio Laboratory, and another 240 feet away from the Laboratory. It was found that when the counterpoise toward the Laboratory was used as a ground, the received current was 342 microampères, but when the counterpoise away from the Laboratory was used, the received current was only 243 microampères. When connected to both counterpoises the received current was 295 microampères, the same as was observed with the net ground.

5.—QUANTITATIVE EXPERIMENTS IN RADIOTELEGRAPHIC TRANSMISSION.

The quantitative study of long-distance radiotelegraphic transmission was begun by the Navy Department in 1909–10 and continued in 1912 in connection with the testing of the high-power radio station at Arlington, Virginia. This station is equipped with a 100-kw. Fessenden rotary-gap sending set which gives an antenna current of approximately 100 ampères at a wave length of 3,800 meters. The aerial is triangular in shape and suspended between three steel towers, two of which are 450 feet in height while the third has a height of 600 feet. The capacity of the antenna is 0.01 m.f. and the height to the center of capacity 400 feet. Short range experiments showed that the effective height of the Arlington station was only about one-half the height to the center of capacity of the antenna. This appears to be generally true of land stations and is probably due to the fact that they are not

erected on sufficiently good conducting surfaces as assumed in the theory. The main scientific object of the experiments was the determination of the correctness of the Sommerfeld transmission formula for received current:

$$(1) \qquad I_R = 120\pi \frac{h_1 h_2 I_S}{\lambda d R} E^{-\frac{0.0019d}{\sqrt[3]{\lambda}}}.$$

where h_1 is the effective height of the sending antenna, h_2 the corresponding height of the receiving antenna, I_S the sending antenna current, λ the wave length, d the distance between the two stations, and R the effective high-frequency resistance of the receiving-antenna system.

The strength of the received signals was measured on the U. S. S. *Salem,* which made a voyage to Gibraltar and return for the carrying out of the tests. The total height of the *Salem's* antenna was 130 feet and the height of the center of capacity 114 feet. The effective high frequency resistance was 50 ohms at 3,800 meters. Signals were received by means of an electrolytic detector and their intensity was measured by the shunted telephone method which was described in the paper already cited. From the data thus obtained it was possible to determine the received antenna current I_R.

Table IV shows the results. Column five gives the experimental values as obtained from the smoothed curve of observations, and column three the values as calculated from the Sommerfeld formula (1). Column four gives the calculated values as obtained from a semi-empirical formula (2) made up of the first term of the theoretical formula but with the absorption term replaced by the absorption term which was found to be correct in the experiments made in 1910. The values in column four are seen to be in very fair agreement with the observed values.

$$(2) \qquad I_R = 120\pi \frac{h_1 h_2 I_S}{\lambda d R} E^{-\frac{0.0015d}{\sqrt{\lambda}}}.$$

The Sommerfeld theory takes no account of energy which may be brought to the receiving station by means of reflection or refraction in the upper atmosphere, and it is thought probable that it is this portion of the energy which produces the difference between the observed and theoretical results.

TABLE IV.

Arlington Received on the "Salem," February–March, 1913.

Resistance.		Received current 10^{-6} ampères.		
		Calculated.		
Miles.	Km.	Eq. 1.	Eq. 2.	Obs.
300	556	335.000	431.00	410.0
400	740	200.000	278.00	300.0
500	925	128.000	195.00	225.0
600	1,110	85.200	140.00	160.0
800	1,480	40.700	79.00	95.0
1,000	1,850	20.700	47.60	59.0
1,200	2,220	11.000	29.70	34.0
1,500	2,780	4.420	15.30	19.0
2,000	3,700	1.070	5.65	5.0
2,500	4,630	0.280	2.20	......
3,000	5,560	0.074	0.84	......

6.—THE TRANSMISSION EFFICIENCY OF CONTINUOUS AND DAMPED WAVES.

An extensive series of experiments have been carried on by the Laboratory on the transmission of damped and undamped waves. It was found that for distances up to 1,000 miles the received antenna currents were practically the same in the two cases for the same sending antenna current. During the voyage of the *Salem* comparison was made between the received day signals from the Fessenden spark set and those from a Poulsen arc temporarily installed at Arlington. For distances above 1,200 miles the arc seemed to be superior for the same sending current. These results were verified later

by receiving the two types of radiation at the naval radio station at Colon. More recently observations have been made on the signals from the 200-kw. spark set at Nauen, Germany, and the 200-kw. high-frequency machine installed in the same station. The wave length in both cases is 9,400 meters. The current with the machine probably varies from 130 to 150 ampères, while the spark antenna currents probably are somewhat higher. At Tuckerton, N. J., observations show that the continuous oscillations are very much stronger, while at the Naval Radio Laboratory the spark signals have never been heard while the machine signals vary between 100 and 5,000 audibility.

7.—A STUDY OF THE ULTRAUDION CIRCUITS.

During the past year much of the time of the Laboratory has been devoted to the development of the DeForest ultraudion. This instrument is a modification of the earlier type

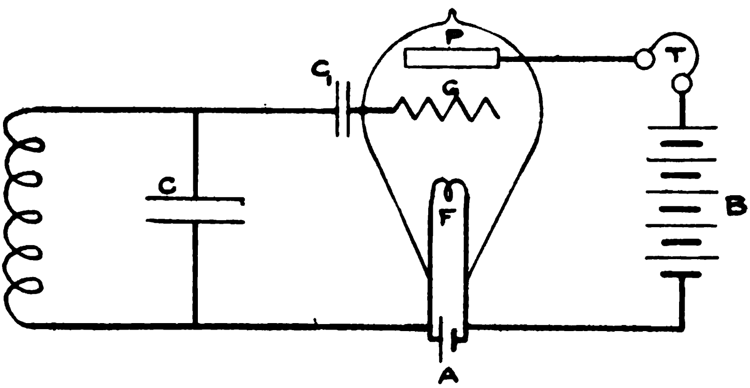


FIG. 3.

of DeForest audion detector, and differs from it only in the method of connecting the oscillatory circuit to the detector. Figure 3 shows the old or filament connection, while Figure 4 shows the ultraudion or plate connection. Under proper conditions of capacity, inductance, resistance, etc., continuous

oscillations* are set up in the circuit connected to the ultraudion in Figure 4. If these oscillations do not take place the

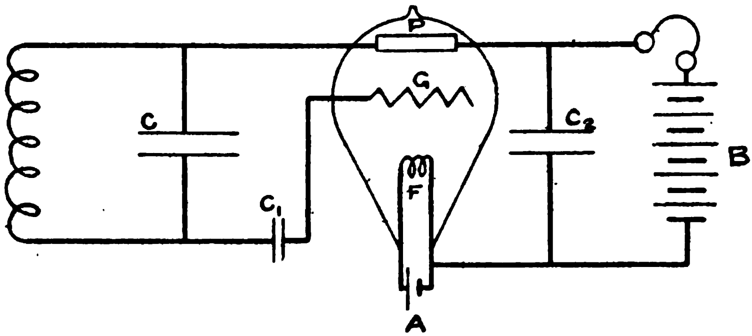


FIG. 4.

ultraudion behaves much like an ordinary audion except that it is many times more sensitive. Undamped oscillations are not heard and spark signals produce the natural note of the sending spark. If oscillations are produced by the audion in the connected circuit, by slightly detuning this circuit from the incoming continuous oscillations, musical beats are produced between the local and incoming oscillations, as in the case of the Fessenden heterodyne. The pitch of the beat tone is entirely determined by the degree of detuning. If the audion is allowed to oscillate when receiving spark signals regular beats cannot be produced and the spark note is roughened.

With the simple circuit without the condenser C_2, Figure 4, oscillations in general do not take place with the capacity C greater than about 0.0005 m.f. and the oscillations are very easily destroyed if any considerable amount of energy is taken from the circuit or if strong atmospheric disturbances impinge upon it.

The main improvements have been along the line of giving greater stability to the oscillations and increasing the sensitiveness. It has been found by Chief Electrician Eaton, assistant

*The presence of these oscillations can be detected by touching with the finger any part of the oscillatory circuit on the grid side of the inductance. If oscillations are present a noise will be heard in the telephones.

in the Laboratory, that the stability of oscillations can be much increased by placing a small condenser across from the grid G (Figure 4) to the plate P and by connecting both grid and plate to the filament by pencil-mark resistances. He also placed a variable air condenser across the telephones, producing a marked increase in sensitiveness. It was afterwards found that both of the auxiliary condensers mentioned could be replaced by one variable condenser C_2, Fig. 4, of 0.002 m.f. connecting the plate and filament. The circuit as now used at the Naval Laboratory is shown in Figure 5. Here the

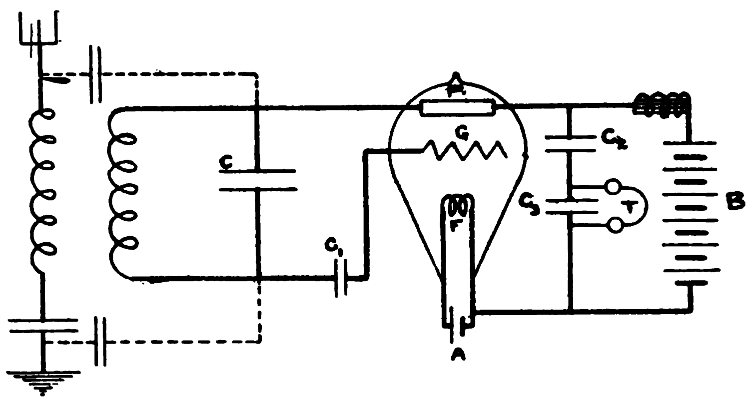


FIG. 5.

original telephones of Figure 4 are replaced by a choke coil and the telephones placed in the shunt circuit P F. The condenser C_2 should not be less in capacity than 0.01 m.f. for full strength of signal, and the condenser C_3 is variable from 0 to 0.002 m.f. By making C_2 variable certain types of atmospherics can be much reduced by decreasing its capacity. With this form of circuit it is possible to use a capacity of somewhat more than 0.002 m.f. for the tuning condenser C of the secondary circuit. The best results are obtained, however, with a capacity not greater than 0.001 m.f.

The stability of the oscillations can be increased and the sensitiveness improved by making use of a static coupling of about 0.0001 m.f. between the primary and secondary cir-

cuits, in connection with the magnetic coupling. The condenser connections for the static coupling are shown by dotted lines in Figure 5. For many purposes only the lower condenser, the one connecting the grid side of the secondary circuit to the ground, is necessary.

8.—PREVENTION OF ATMOSPHERIC DISTURBANCES.

The most important practical unsolved problem in radio reception is undoubtedly the prevention of atmospheric disturbances, for it is this, and this alone, which keeps radio communication from attaining the same certainty of operation as that attained by wire. For this reason a vast amount of time has been spent in the Laboratory in testing every device which has been proposed by radio experimenters or which has been thought of by the Laboratory staff. While most of these devices have proved useless, a few have given some degree of success.

One of the best methods so far tried in the case of ordinary receiving sets with contact detectors is that of placing a contact composed of silicon and arsenic around the receiving antenna inductance. Contacts of this character have the property of offering a very high resistance to small electromotive forces, and a comparatively low resistance to electromotive forces of any considerable magnitude. In this way strong atmospheric disturbances and powerful interfering signals may be partly shunted to earth, while the weaker received signals pass through the inductance with practically no diminution in intensity. The general character of the action is shown in Table V, the deflections being those of a galvanometer attached to the regular receiving detector, with the disturbance-preventing circuit connected and disconnected. The antenna was excited by a tuned buzzer circuit with different degrees of coupling. It is seen that the weak signals such as ordinarily are received are not at all weakened by the disturbance preventer, while the strong signals are cut down to less than a tenth of their full value.

TABLE V.

Disturbance Preventer.

In. Deflection. mm.	Out. Deflection. mm.
0.5	0.5
1.5	1.5
3.5	4.5
7.0	22.0
15.0	130.0
21.0	240.0
40.0	off scale
2.5	2.5

This device has been used to a considerable extent in receiving in various radio stations. Its chief practical drawback seems to be the difficulty of getting correct adjustment of the contact, this being considerably more critical than the ordinary detector adjustment. It is also found that in the case of very large antennas the atmospheric strokes are frequently so powerful that the disturbance preventer contact is broken down, thus limiting the applicability of the device in large stations.

For the best results the antenna should contain considerable inductance with a series condenser if necessary. The coupling should be loose.

In the new audion circuits the prevention of atmospheric disturbances is a somewhat different problem from that met in the old-type circuits. The use of loose coupling is less advantageous and appears to result in a diminution in strength of signal quite as rapid as the diminution of the atmospherics.

It is extremely fortunate that the oscillating audion is far less sensitive to disturbances than to the signals, but nevertheless atmospherics can be extremely troublesome and thus far no means have been found for controlling them. With certain types of disturbance considerable relief can be obtained

by reducing the capacity of the condenser C_2 in Figure 5. The connection of the telephones through a tuned variable coupling iron core telephone transformer, as shown in Figure 6, gives further improvement. In this the primary and secondary may have approximately the same inductance: *i. e.*, about 200 milli-

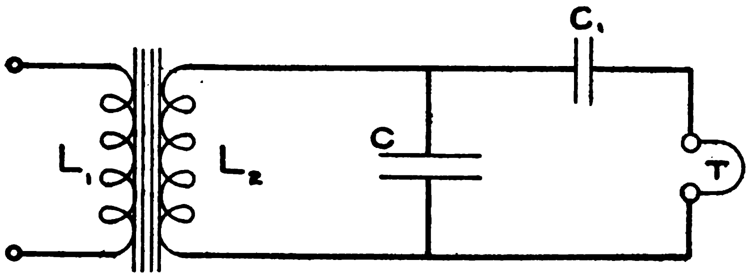


FIG. 6.

henries, while C is about 0.01 m.f. and C_1 is 0.01 or 0.02 m.f. By making both C and C_1 variable the transformer can be tuned to any tone desired. The primary of the transformer is inserted in the place of the telephones as shown in Figure 3.

Below is given a list of the papers published by the Laboratory since the last report:

1. Antenna Resistance. Bulletin of the Bureau of Standards, IX, p. 65, 1912. Reprint 189, and Jahrbuch d. drahtlosen telegraphie, V, p. 574, 1912.

2. Energy Losses in Some Condensers Used in High-Frequency Circuits. Bulletin of the Bureau of Standards, IX, p. 73, 1912. Reprint 190, and Jahrbuch d. drahtlosen telegraphie, VII, p. 222, 1913.

3. Comparison of Damped and Undamped Oscillations. Journal of the Washington Academy, II, p. 111, 1912, and Jahrbuch d. drahtlosen telegraphie, V, p. 524, 1912.

4. Suitable Wire Sizes for High-Frequency Resistance. Journal of the Washington Academy, II, p. 112, 1912, and Jahrbuch d. drahtlosen telegraphie, VI, p. 588, 1912.

5. The Relation Between Effective Resistance and Frequency in Radio Telegraphic Condensers. Proc. Institute of Radio Engineers, I, p. 35, 1913.

6. The High-Frequency Resistance of Inductances, Journal of the Washington Academy, III, p. 94, 1913, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 159, 1914.

7. The Measurement of Received Radiotelegraphic Signals. Journal

of the Washington Academy, III, p. 133, 1913, and Jahrbuch d. drahtlosen telegraphie, VII, p. 628, 1913.

8. A Comparison of Arc and Spark Sending Apparatus. Journal of the Washington Academy, III, p. 284, 1913, and Jahrbuch d. drahtlosen telegraphie, VII, p. 506, 1913.

9. Difference in Strength of Day and Night Signals. Journal of the Washington Academy, III, p. 326, 1913, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 381, 1914.

10. A Crystal Contact Disturbance Preventer for Radio Telegraphic Receiving. Journal of the Washington Academy, III, p. 386, 1913, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 481, 1914.

11. The Effect of a Parallel Condenser in the Receiving Antenna. Pros. Inst. of Radio Engineers, II, p. 131, 1914, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 524, 1914.

12. Quantitative Experiments in Radio Telegraphic Transmission, Bulletin of the Bureau of Standards, XI, p. 69, 1914, Reprint 226, and Jahrbuch d. drahtlosen telegraphie, VIII, p. 575, 1914.

POSSIBLE APPLICATION OF THE DRZEWIECKI METHOD TO THE DESIGN OF WATER PROPELLERS.

By Assistant Naval Constructor H. E. Rossell, U. S. N., Member.

The remarkably high efficiencies obtained with air propellers designed by the Drzewiecki or constant incidence method have caused much speculation as to the applicability of this method of design to water propellers. The best air propellers at present have efficiencies as high as 85 per cent., while the highest obtained with water propellers is about 75 per cent.

As is well known, the conventional methods of designing water propellers all consider the propeller to be analogous to a nut working on a thread. The distance the propeller would move along its axis in one revolution, if it were working on a rigid thread, is called the true pitch. However, in one revolution, the propeller actually moves through the water in which it is working an axial distance less than the true pitch. This distance is ordinarily called the effective pitch, the percentage difference between true and effective pitch being called the real slip. The speed of the wake of the ship, in which the propeller works, is assumed to be some percentage of the speed of the ship. In designing a propeller, the speed of advance through the water, the thrust which must be developed, and the revolutions are usually fixed. The diameter can generally lie within certain rather narrow limits. The design variables are, therefore, the pitch and the blade width. Chief Constructor Taylor's method is based on the performance of model propellers, as is Professor Peabody's, while Captain Dyson's method is based on the performance of full-size pro-

pellers. Any of these methods will give a propeller which will do the work at an efficiency of from 65 to 70 per cent. With few exceptions all water propellers have a helicoidal after surface, the thickness all being on the forward side or back of the blades. A plane section of a blade, parallel to the axis of a propeller, usually has its maximum ordinate at the center of the chord and is symmetrical with respect to this ordinate.

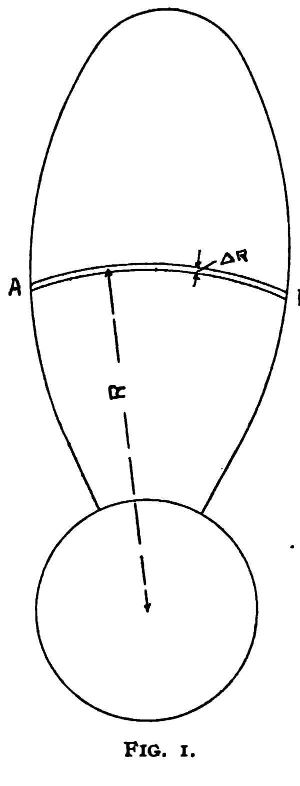


FIG. 1.

In the Drzewiecki method of design any section of the blade, as AB, Fig. 1, is considered as a section of a wing, having a velocity V_t through the medium in which it is working. V_t is the velocity compounded from the speed of rotation at radius R, and the speed of advance through the water. The curvature of the section, AB, is disregarded, and it is assumed

to act as if it were a plane section moving at a velocity, V_t, with respect to the air, or water. The forces, acting on the section, may be resolved into lift component, $\triangleleft L$, and drift component, $\triangleleft D$, respectively perpendicular to and along the direction of motion. The angle, cot $= {}^1\triangleleft L \div \triangleleft D$, we will call G; and the angle between the direction of motion and

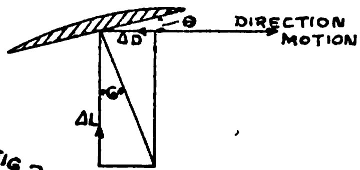


FIG 2.

the chord of the section we will call θ, Fig. 2. Angle A, Fig. 3, is the angle between the direction of motion and the normal to the axis of the propeller.

Force along axis $= \triangleleft L \cos A - \triangleleft D \sin A$.
Force perpendicular to axis $= \triangleleft L \sin A + \triangleleft D \cos A$.

$$\text{Efficiency of element} = \frac{(\triangleleft L \cos A - \triangleleft D \sin A) \sin A}{(\triangleleft L \sin A + \triangleleft D \cos A) \cos A}$$

$$= \frac{(\cos G \cos A - \sin G \sin A) \sin A}{(\cos G \sin A + \sin G \cos A) \cos A}$$

$$= \frac{\cot (A + G)}{\cot A}$$

$\cot A = 2\pi R \div P_e$, where P_e is the effective pitch.

Now, if the lift per unit area and the ratio of lift to drift at each section are known, the forces acting along the axis at all parts of the blade are known, and can be summed up to get the thrust.* A large number of tests have been made in air on model wings of various sections. These tests usually give the ratio of lift to drift and the lift, either in the form of a coefficient or directly, for various angles of incidence. By

* For an excellent description of the method used in actually making the design, see Report No. 65, by Mervyn O'Gorman, Technical Report of the Advisory Committee for Aeronautics, 1911-1912.

angle of incidence is meant the angle between the chord of the wing and the direction of motion. It is the angle θ, in Fig. 3. Fig. 5 shows the lift ÷ drift and lift coefficient curves

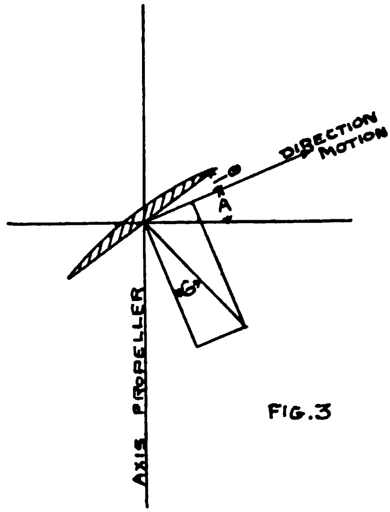


FIG. 3

for two wings, selected from the Technical Report of the Advisory Committee on Aeronautics, London, 1912–1913. These curves show that each wing has a definite angle of incidence at which the ratio of lift to drift is a maximum. For wing No. 0 in Fig. 5 this incidence is 8 degrees; while for wing No. 2 it is 4 degrees. Maximum ratio of lift to drift gives a minimum gliding angle G, and maximum efficiency for any section of the propeller having the most advantageous angle of incidence. It is, therefore, the aim, in this method of design, to give the propeller blade such a form as to cause each section to operate at the angle of incidence for which the ratio of lift to drift is a maximum. In other words, the angle θ, Fig. 3, is kept the same for all sections of the blade.

In selecting a suitable section for a propeller the character of the lift curve is of importance. This section adopted must have a good lift at the incidence used in order to get the desired thrust with reasonable blade width and propeller diameter.

In order to get some idea of the gain in efficiency to be expected from giving the propeller constant incidence, let us consider a water propeller, designed by Professor Peabody's method, and having the following specifications.

Diameter, 6.87 feet.

Pitch, 7.90 feet.

Efficiency, 68 per cent.

Projected area, 10.01 square feet.

Peripheral velocity, 4,854 feet per minute.

Developed area, 12.21 square feet.

Number of blades, 3.

Width ratio, .2184.

True slip, 25 per cent.

Speed of ship, 14.5 knots.

I.H.P. (each engine), 730.

Revolutions, 225.

Wake factor (assumed), .1.

Speed of wake $(.9 \times 14.5 \times 6{,}080) \div 60 = 1{,}322$ feet per minute.

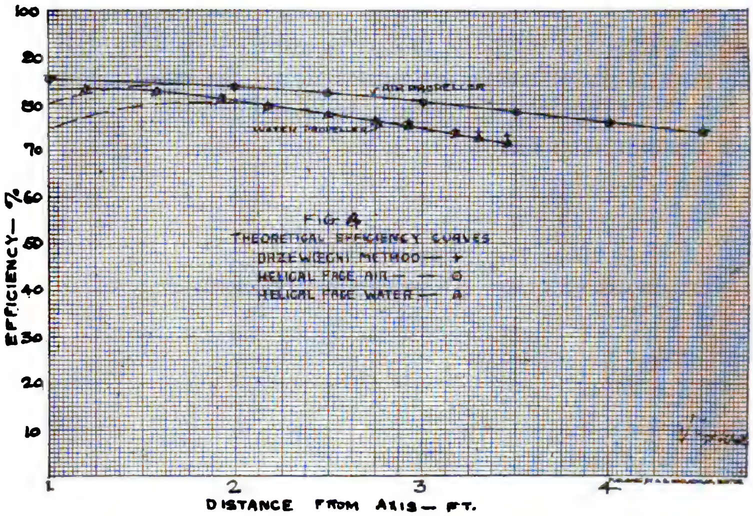
Effective pitch, $P_e = 1{,}322 \div 225 = 5.877$ feet.

$\cot (A - \theta) = 2\pi R \div P$, where P is the true pitch.

Solving for θ, we find that it varies from 9.2 degrees at radius of 10 inches to 4.9 degrees at the blade tips. For the sake of comparison, let us assume that the section of the propeller is that of wing No. 0 in Fig. 5, and that the curve of lift $\div$ drift is the same in water as in air. The ratio of lift to drift may be taken from the curve for each angle of incidence, and the theoretical efficiency curve for the helical propeller worked out. This curve is shown in Fig. 4.

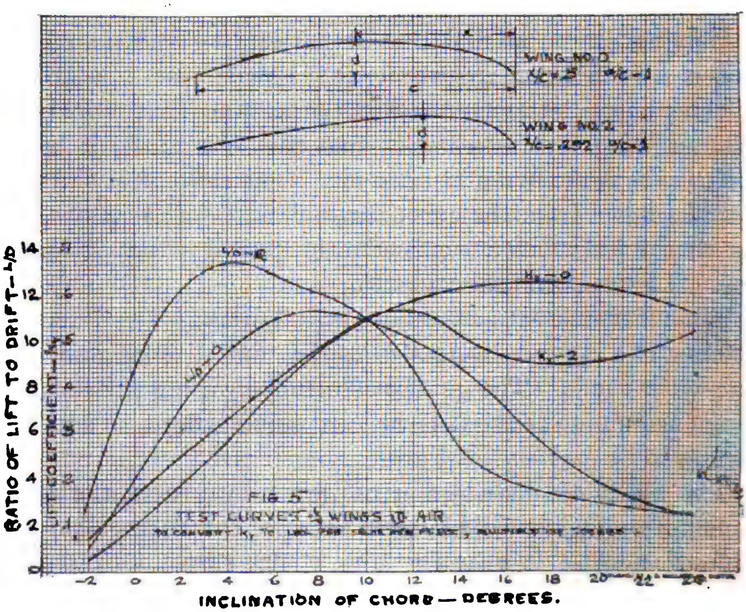
The efficiencies at various sections, assuming a constant incidence of 8 degrees, were also worked out, and are shown by crosses in Fig. 4. The gain in efficiency by using the constant incidence method is seen to be very small indeed, so small in fact as to be entirely negligible.

The effect on distribution of thrust by giving the blade constant incidence would be to increase the thrust near the tip and decrease it near the hub. The thrust in this case



would be increased at the blade tip about 10 per cent. This would probably cause a greater tendency toward cavitation.

Now suppose that we have adopted section No. 2 for an air propeller with the following specifications:



Diameter, 9 feet.
Slip, 25 per cent.
Revolutions, 1,200.
Speed, 90 miles per hour.

The theoretical efficiency curve assuming a helicoidal surface is shown in Fig. 4. The efficiencies for constant incidence are, as before, designated by crosses. Actually the high efficiency cannot be maintained near the hub, where the sections must be thickened in order to get the necessary strength. The probable drooping of the efficiency curves is shown in dotted lines in Fig. 4. The gain in efficiency by using the constant incidence method of design is seen to be even smaller than in the case of the water propeller.

CONCLUSION.

The Drzewiecki method is well adapted to the design of air propellers. By using it and the published tests on wing sections, one can design a propeller for any air craft, and be confident of getting very high efficiency and of developing the required torque at the designed revolutions. There is, however, no ground for the belief that the high efficiency is obtained by using constant incidence. The gain from this source is of the order of .2 of one per cent. The great advance made in air propeller efficiency is attributable to the more efficient sections which have been discovered.

There seems to be no advantage in applying this method to the design of water propellers. The helical propeller is cheaper to make of metal than the Drzewiecki propeller. The conventional methods of design are much easier to use than the constant incidence method, and give as good efficiencies. Moreover there are no tests available of wings in a current of water. The tests of wings in air cannot be used with any certainty, because the frictional and eddy-current losses are probably much larger in water than in air. The method does suggest a very rational way of investigating the efficiencies

of various sections which might be used for water propeller blades. It might, for instance, be much easier to test a given section for lift and drift in water than to determine, by test, the efficiency of a propeller built with this section. It is now a recognized fact that the form of the back of the blade section has considerable influence on the performance of either an air or water propeller. If any marked advance is to be made in the efficiencies of water propellers it must be from the discovery of a more efficient blade section over those which have been tried heretofore.

U. S. S. *BALCH.*

CONTRACT-TRIAL PERFORMANCE.

By Henderson B. Gregory, Associate.

Torpedo-boat destroyer No. 50, the *Balch,* is one of eight destroyers authorized by an Act of Congress, approved March 4, 1911. She is a sister ship of the *Aylwin, Parker* and *Benham,* the four vessels being built under contract by the William Cramp and Sons Ship and Engine Building Company of Philadelphia, Pa., at a price of $756,100.00 each. The time of completion for the *Balch* was twenty-four months from the date of signing the contract, September 7, 1911. She was designed for a speed of 29.5 knots, at about 1,036 tons displacement, with the main engines developing 16,000 shaft horsepower.

The *Balch* is a twin-screw vessel fitted with a combination of Cramp-Zoelly turbines and reciprocating engines, one of each on each shaft. For speeds of 17 knots and above the turbines alone are used, and for the lower cruising speeds the reciprocating engines are used in combination with the turbines, disconnecting couplings being installed for readily throwing the latter in and out of gear. Steam is supplied by four oil-burning White-Forster water-tube boilers, arranged in pairs in two separate compartments.

The hull and machinery are duplicates of the *Aylwin, Parker* and *Benham,* a complete description of which was published in the February, 1914, number of the Journal of the American Society of Naval Engineers, and which likewise applies to this vessel.

PRINCIPAL HULL AND MACHINERY DATA.

Hull dimensions:	
Length on L.W.L., feet and inches	300–0
over all, feet and inches	305–3
Breadth at L.W.L., feet and inches	30–4
Ratio, breadth to length (L.W.L.)	10.064
Draught to L.W.L., feet and inches	9–2½
Displacement corresponding, tons	1,010
per inch at L.W.L., tons	14.21
Area immersed midship section, square feet	190
Coefficient of fineness, block	0.415
Main turbines:	
Working pressure at chest, pounds gage	240
R.P.M., designed	620
S.H.P., designed	16,000
Number of stages, ahead	16
astern	8
Cruising engine:	
Type	Vertical, inverted, direct-acting, compound, jacketed.
Cylinder, H.P., diameter, inches	13
L.P., diameter, inches	25
Stroke, inches	12
Crank angle, degrees	180
R.P.M. (corresponding to a speed between 15½ and 16 knots)	300
I.H.P. (corresponding to a speed between 15½ and 16 knots)	460
Boilers:	
Type	White-Forster, oil burning, water tube.
Working pressure	260
Number	4
Furnace volume, each, cubic feet	452
Number of smoke pipes	4
Area through each smoke pipe, square feet	16.57
$\frac{\text{Area through smoke pipe}}{\text{Furnace volume}}$	0.03666
Heating surface, each, square feet	5,400
$\frac{\text{Heating surface}}{\text{Furnace volume}}$	11.9
Oil burners, number each boiler	11
type	Schutte-Koerting.
Diameter, inches, tubes (outside)	1
downcomers (inside)	8⅛
steam drum (inside)	48
lower drums (inside)	19½

TRIALS.

The following trials were required by the contract:

(*a*) A progressive trial over a measured-mile course for standardizing the screws, extending from maximum speed down to a speed of twelve knots.

(*b*) A full-speed trial of four hours' duration in the open sea, in deep water, at the highest speed attainable, the average speed to be not less than 29.5 knots, with the average air pressure in the firerooms not exceeding 6 inches of water and the steam pressure at the high-pressure turbine not to exceed 240 pounds above the atmosphere.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea at an average uniform speed of 24 knots as nearly as possible.

(*d*) A fuel-oil and water-consumption trial of four hours' duration in the open sea at an average speed of 15½ knots as nearly as possible. This trial to be made as nearly as possible under service conditions, with the cruising engines connected up and in use.

(*e*) An endurance trial of twenty hours' duration in the open sea at an average speed of 15½ knots as nearly as possible, under service cruising conditions, with cruising engines connected up and in use.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea with the cruising engines connected up and in use, at an average speed of 12 knots as nearly as possible.

Fuel Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 705 pounds at the guaranteed maximum speed of 29.5 knots, 445 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots. For the purpose of ascertaining the fuel-oil consumption the fuel oil burned on trials (*b*), (*c*), (*d*) and (*f*) was carefully measured, from which the consumption at the stipulated speeds was computed

TABLE I—STANDARDIZATION TRIAL DATA—FEB. 21 & 22, 1914—U.S.S. BALCH

No. of Run	Time on Course: Mins.	Time on Course: Secs.	Speed in Knots	Revs. to go One Knot: Star. Engine	Revs. to go One Knot: Port Engine	Revs. to go One Knot: Mean	R.P.M.: Star. Engine	R.P.M.: Port Engine	R.P.M.: Mean	S.H.P.: Star. Engine	S.H.P.: Port Engine	S.H.P.: Total
1	4	11.1	14.337	1042.4	1045.8	1044.1	248.75	249.56	249.15	751	681	1432
2	3	50.3	15.632	961.6	959.8	960.7	250.22	249.75	249.99	771	697	1468
3	4	4.3	14.736	1009.7	1010.2	1000.0	248.00	248.10	248.05	734	705	1439
Mean of Group			15.084						249.30			1452
4	3	14.3	18.528	1013.3	1015.3	1014.3	312.94	313.56	313.25	1418	1380	2798
5	3	12.7	18.682	1002.5	997.4	1000.0	312.05	310.46	311.25	1429	1385	2814
6	3	12.3	18.721	998.0	1000.9	999.5	311.39	312.29	311.84	1445	1358	2803
Mean of Group			18.654						311.90			2807
7	2	45.7	21.726	1009.9	1011.8	1010.9	365.23	366.23	365.73	2373	2205	4578
8	2	46.2	21.661	1021.5	1026.6	1024.1	368.84	370.66	369.75	2434	2331	4765
9	2	43.1	22.072	999.9	1006.5	1003.2	367.73	370.18	368.95	2405	2373	4778
Mean of Group			21.780						368.55			4722
10	2	32.0	23.684	1062.8	1060.9	1061.9	419.15	418.42	418.78	3818	3611	7429
11	2	27.0	24.490	1025.6	1025.5	1025.6	418.27	418.23	418.25	3693	3517	7210
12	2	32.4	23.622	1071.7	1070.0	1070.9	422.10	421.43	421.76	3871	3734	7605
Mean of Group			24.072						419.26			7364
13	2	11.1	27.460	1089.1	1085.8	1087.5	498.22	496.71	497.46	5989	5812	11801
14	2	17.4	26.201	1141.8	1142.2	1142.0	498.60	498.76	498.68	6327	6085	12412
15	2	9.8	27.735	1083.7	1075.8	1079.8	500.77	497.12	498.94	6159	6095	12254
Mean of Group			26.900						498.44			12220
19	4	35.5	13.067	885.4	882.9	884.2	193.53	192.99	193.26	358	322	680
20	5	32.8	10.817	1058.7	1074.0	1066.4	190.98	193.74	192.36	321	335	656
21	4	51.8	12.337	935.6	935.5	935.6	191.97	191.95	191.96	344	299	643
Mean of Group			11.760						192.50			659
23	2	2.9	29.292	1204.7	1197.4	1201.1	588.37	584.81	586.59	8455	—	—
24	2	2.5	29.388	1189.3	1188.1	1188.7	583.28	582.69	582.98	8382	8472	16854
25	2	5.2	28.754	1220.0	1206.3	1213.2	584.41	577.85	581.13	8363	8078	16441
Mean of Group			29.206						583.42			16648
26	1	59.6	30.100	1189.1	1180.3	1184.7	597.52	593.07	595.29	8784	8632	17407
27	2	4.4	28.939	1257.4	1244.4	1250.9	606.27	599.98	603.12	8912	9024	17936
28	1	57.0	30.769	1175.2	1166.5	1170.9	603.76	599.26	601.51	9014	8749	17763
29	2	5.6	28.662	1276.3	1261.0	1268.7	609.94	602.60	606.27	9135	8931	18066
30	1	54.5	31.441	1151.8	1146.9	1149.4	603.80	601.26	602.53	9015	8941	17956
[illegible]			[illegible]						[illegible]			17862

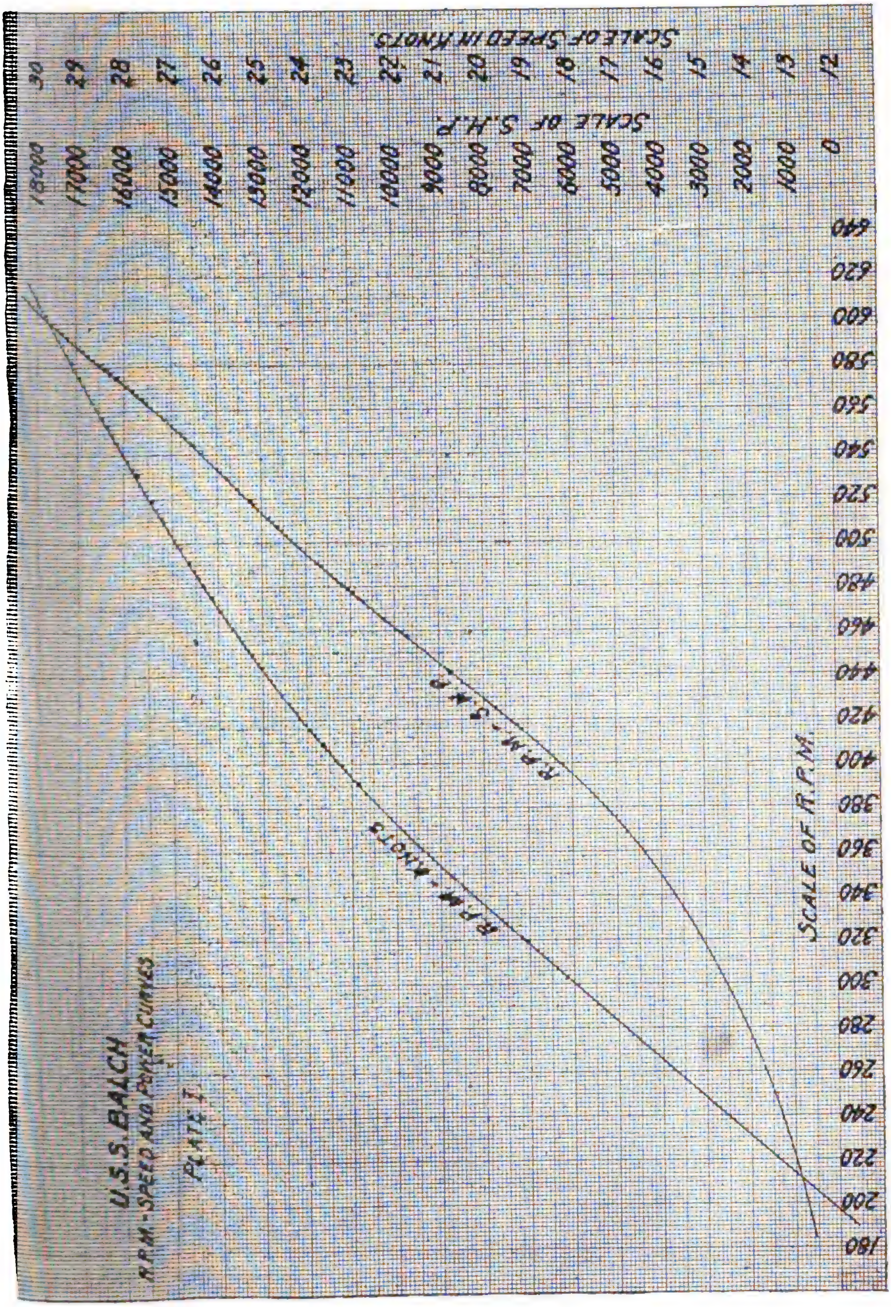

U.S.S. BALCH
R.P.M.-SPEED AND POWER CURVES
PLATE I
SCALE OF SPEED IN KNOTS.
SCALE OF S.H.P.
SCALE OF R.P.M.
R.P.M. - S.H.P.
R.P.M. - KNOTS

TABLE II - TRIAL DATA - U.S.S. BALCH.

	4-Hrs.' Full Power Trial (b)		4-Hrs.' 24-Kt. Trial. (c)		4-Hrs.' 15½-Kt. Trial. (d)		4-Hrs.' 12-Kt. Trial. (f)	
Date of Trial	Feb. 26, 1914.		Feb. 22, 1914		Feb. 27, 1914.		Feb. 23, 1914	
Speed in Knots	29.618		24.031		15.594		12.208	
Draught, mean on trial, feet and inches	9 - 4 3/8		9 - 6 1/2		9 - 5 3/8		9 - 7 11/32	
Displacement, corresponding, tons	1023.9		1053.0		1050		1065.5	
Slip of Propellers, percent, starb'd	24.59		14.51		7.87		7.06	
port	24.62		12.83		8.63		7.65	
mean	24.605		13.67		8.25		7.355	
Engines in operation	Main Turbines		Main Turbines		Main Turb. & Cruising Eng.		Main Turb. & Cruising Eng.	
Number of Boilers used	4		4		2		2	
Burners used (11 per Boiler) (Average)	41.5		21 3/8		5		3 3/8	
Heating Surface used, sq. ft.	21,600		21,600		10,800		10,800	
sq. ft. per S.H.P.	1.2521		2.9095		6.805		15.697	
Cooling Surface, sq. ft. (Main Cond.) per S.H.P.	0.6957		1.6166		7.5626		17.445	
Pressures (Average):	Stb'd.	Port	Stb'd.	Port	Stb'd.	Port	Stb'd.	Port
Main Steam, at Boilers, pounds gauge	249		248.1		245		242.2	
Eng. Room, " "	246.8	249.1	245.4	247.0	244.4	246.6	245.6	246.4
Full Speed Ahead Chest, " absolute	215.0	215.0	84.4	87.9	21.6	15.0	14.1	13.6
Medium " " " " "	192.4	202.6	212.5	213.8	36.6	29.4	25.3	23.5
10th Stage, pounds absolute	31.7	32.1	17.1	17.0	3.9	3.8	1.0	1.0
Gland Steam, " gauge	6.8	9.8	10.0	7.6	3.6	3.0	2.4	3.0
H.P. Valve Chest, Recp. Eng., pounds gauge	—	—	—	—	223.6	214.4	150.0	151.1
L.P. Exhaust, " " " absolute	—	—	—	—	41.3	40.3	26.6	31.1
Vacuum, Main Cond., inches of mercury	28.9	29.03	29.00	29.30	28.84	29.6	29.0	29.8
Barometer, inches of mercury	30.2		30.02		30.04		30.05	
Auxiliary Steam, pounds gauge	247.3		243.9		241.0		244.3	
Exhaust, " "	6.0		7.9		7.8		3.2	
Main Feed, at Pumps, " "	300.0		304.0		300.0		330.6	
Forced Lubrication, " "	40.9		41.9		60.0		55.0	
Fuel Oil, pounds gauge	211.1		207.5		180.0		214.4	
Air Pressure in Fire Rooms, inches of water	7.27		4.23		2.15		1.3	
Revolutions or Double Strokes per min. (Average)								
Starb'd Shaft	596.96		418.25		257.25		199.65	
Port "	597.17		419.01		259.37		200.93	
Mean	597.06		418.63		258.31		200.29	
Main Air Pumps	18.5	16.6	21.3	22.3	—	16.5	—	17.0

REVOLUTIONS OR DOUBLE STROKES PER MIN. (AVERAGE)				
FEED PUMPS	(2) 28.9	(2) 18.15	(1) 8.8	(1) 5.4
FORCED DRAFT BLOWERS	(4) 1,662.0	(4) 1205.0	(1) 918.1	(1) 676.3
FUEL OIL SERVICE PUMPS	(2) 25.1	(2) 10.75	(1) 4.0	(1) 3.3
SHAFT HORSE POWER (AVERAGE)			TOTAL S.H.P. | I.H.P. CR. ENG.	TOTAL S.H.P. | I.H.P. CR. ENG.
STARB'D SHAFT	8,716	3,712	826 | 409	364 | 220
PORT "	8,535	—	761 | 395	324 | 203
TOTAL	17,251	7,424 *	1587 | 804	688 | 423
TEMPERATURES, DEGREES F. (AVERAGE)				
MAIN FEED TANK	70.6	51.5	50.9	50.4
INJECTION	45.7	42.9	36.5	32.0
OUTBOARD DELIVERY	74.7 | 75.3	58.9 | 58.8	42.8 | 42.5	37.3 | 37.1
OUTSIDE AIR	46.0	32.0	38.0	26.0
ENGINE ROOM	73.9	68.5	71.9	84.1
AUXILIARY ROOM	71.6	70.9	69.2	62.3
FIRE ROOMS	68.35	74.7	73.4	66.4
FEED WATER	177.5	190.7	209.1	205.1
FUEL OIL TO HEATERS	45.2	48.1	52.5	48.9
BURNERS	112.7	147.6	163.2	121.9
SMOKE PIPES	504.4	416.6	516.2	301.3
WATER CONSUMPTION				
POUNDS PER HOUR, MEASURED	291,083.70	143,218.4	37,814.9	19,562.1
RESERVE FEED	536.25	834.5	655.0	296.25
TOTAL EVAPORATED	291,619.95	144,052.9	38,469.9	19,858.35
PER S.H.P.	16.874	19.292	23.828	28.433
EVAP'D PER SQ. FT. OF H.S.	13.5	6.67	3.56	1.84
LB. OF OIL	13.042	13.926	15.516	12.792
KNOT RUN	9,846.04	5,994.46	2,466.97	1,626.67
FUEL OIL CONSUMPTION				
B.T.U. PER POUND	19,158	19,158	19,158	19,158
SP. G. AT 60° F.	0.905	0.905	0.905	0.905
POUNDS PER HOUR	22,318.2	10,284.79	2,437.2	1,529.23
KNOT RUN	753.54	427.98	156.29	125.26
AT	29.5 KTS. 745.0	24 KTS. 426.5	15.5 KTS. 155.0	12 KTS. 124.0
GUARANTEED, AT	705.0	445.0	213.0	170.0
BELOW (B) OR ABOVE (A) GUARANTEE	(A) 40.0	(B) 18.5	(B) 58.0	(B) 46.0
HOUR PER S.H.P.	1.2937	1.3853	1.5357	2.2227
SQ. FT. OF H.S.	1.033	0.476	0.226	0.142

* DOUBLED STARB'D. SHAFT. PORT TORSIONMETER OUT OF ORDER.

The water consumption was also carefully measured on these trials.

Trial (*a*).—The standardization trial was run on the measured-mile course off Delaware Breakwater. It was commenced at 2:00 P. M., February 21, 1914, and temporarily discontinued at 5:35 P. M., on account of darkness, eighteen runs having been made. The trial was resumed at 7:00 A. M. the following day and was completed at 9:00 A. M., twelve additional runs being made, Nos. 19 to 30, inclusive. The conditions of weather and sea were favorable throughout the runs.

The data obtained, at the several speeds, is given in Table I, from which the curves, Plate I, were plotted. Runs Nos. 16, 17, 18 and 22 were not used in plotting the curves, conditions being unsatisfactory.

It was deduced from the official revolution-speed curve that the following mean r.p.m. of the two shafts, are required to obtain the speeds noted:

R.p.m.	Knots.
196.9	12
256.7	15.5
417.7	24
593.3	29.5

It was estimated that the displacement at the middle of the five high runs was 1,049 tons, corresponding to a mean draught of 9 feet 6⅛ inches.

Trial (*c*).—The trial was started at 12:35 P. M., shortly after the conclusion of trial (*a*). The weather and sea conditions were fair. It was run in deep water off Five Fathom Bank Lightship and vicinity and was entirely satisfactory. The data obtained are tabulated in Table II.

Trial (*f*).—At 8:50 A. M., February 23, the vessel began the trial, which was run with the cruising engines and turbines in combination. The weather was good and the trial was successfully completed at 12:50 P. M. The trial was con-

ducted in the lower part of Delaware Bay and outside the Breakwater. The data are given in Table II.

Trial (*b*).—This trial, at highest speed attainable, was commenced at 10:35 A. M. on February 26, no trials being held February 24 and 25 on account of bad weather. Conditions were generally favorable. The course was off the coast in a southerly direction between Delaware and Chesapeake Bays. The contract speed of 29.5 knots was slightly exceeded, but the fuel-oil consumption was in excess of the guarantee, as noted in Table II, which also contains the general data of the trial.

Trial (*d*).—Following trial (*b*) the four-hour fuel-oil and water-consumption trial at 15.5 knots, with cruising engines in use, was run off the entrance of Chesapeake Bay, on the afternoon of the same day. The weather conditions were very good and the fuel-oil consumption was readily bettered. For data see Table II.

Trial (*e*).—This trial began at 7:41 P. M., February 26, immediately following trial (*d*). The trial was run outside the entrance of Chesapeake Bay until daylight of the 27th; the remainder of the trial was run in the Bay. The weather conditions were excellent. Throughout the trial the machinery worked well, with few minor exceptions, and the performance was considered satisfactory. The vessel maintained an average of 258.6 r.p.m., corresponding to a speed of 15.61 knots per hour. No other data was taken.

GAS ANALYSIS.

By W. N. Berkeley, Ph. D., Chemist, U. S. Naval Engineering Experiment Station.

The whole rationale of gas analysis depends on the fact that if you bring together, under suitable conditions of temperature, state of subdivision, and of intimate mixture, a combustible material and a supporter of combustion (the masses of these two bearing to one another a certain quantitative relationship) there is formed, as the sole product of combustion, carbon dioxide (CO_2) in the theoretical case of a combustible substance containing carbon alone, or, in addition to these, water (H_2O) in the usual case of a combustible substance containing carbon and hydrogen.

Also, that in the presence of an excess of the "combustible" you change the composition of the products of combustion by the addition of carbon monoxide (CO), frequently in considerable amount, and possibly by the further addition of certain hydrocarbons, which are apt to be produced by the same conditions that give rise to the formation of carbon monoxide.

Likewise, that in the presence of an excess of the supporter of combustion (air), you get with an excess of air such a deficiency of carbon dioxide as to readily reveal the fact that an unnecessary excess of air is being used.

DIFFERENT OPINIONS AS TO WHICH CONSTITUENT OF FLUE GAS SHOULD BE TAKEN AS THE MOST RELIABLE INDEX TO CONDITIONS OF COMBUSTION.

As to which constituent of flue gas should be regarded as the most reliable index to conditions of combustion, there is a considerable difference of opinion, though there seems to be a decided disposition on the part of most authorities to assign this rôle to carbon dioxide.

Hempel ("Gas Analysis," p. 256), however, holds that the conditions of combustion in a furnace are most accurately indicated by the percentage of oxygen in flue gas; while Randall (Bulletin 373, U. S. Geological Survey, page 11) says that carbon monoxide is "in general a good guide to efficient operation."

As has been said, however, an examination of the statements contained both in standard books on gas analysis and in technical and semi-technical journals shows that a preponderance of opinion attributes the dominating influence to carbon dioxide; or, to speak more accurately, that it is the percentage of carbon dioxide that most clearly indicates the character of combustion.

Although this difference of opinion is unfortunate, it should not be thought that it impairs, to any great extent, the unquestioned and unquestionably great value of flue-gas analysis; for, while it may not be possible at the present time to assign to each constituent just exactly the proportionate influence that it exerts on the character of combustion, or rather to determine just what the percentage of each constituent indicates as to the character of combustion, yet there are certain limiting percentages that are assigned to each constituent, as permissible under satisfactory conditions; and it is because flue gas analysis affords a very reliable means of detecting even a slight departure from these limiting values that it is as important an index to the condition of operation of a furnace as is the indicator diagram to that of an engine.

AN OUTLINE OF THE METHOD OF MAKING A FLUE-GAS ANALYSIS.

The process, in brief, consists in measuring accurately any convenient volume of the gas, preferably 100 cubic centimeters, and the passage of this through a series of absorbents, by which various constituents are successively absorbed, the volume of each being determined by the reduction in the volume.

Such determinations are most readily made by means of some form of the well known Orsat type of gas analysis apparatus, such for instance, as is shown in Fig. 1. The details of the analysis, as given below, therefore, will apply especially to this particular form of apparatus; but a thorough understanding of the manipulation of this instrument will permit an easy adjustment of the instructions to any similar instrument.

HOW TO SECURE A REPRESENTATIVE GAS SAMPLE.

No matter how much care and skill are exercised in the analysis of a sample, unless this represents the average composition of the material sampled (or, at least, its composition at the time of sampling), the determination of its composition will be a worse than useless operation, for it will not only furnish any data of value but may lead to entirely erroneous conclusions. In order to secure truly representative samples of flue gas, it is necessary that portions be drawn from a large number of points in the cross-section of the stack, or "pass," *and that there should be a continuous flow of gas through the sampling tube and all connecting tubes between this and the sampler.* In this way small portions are taken from a very large volume (just as in sampling coal, ore, etc., small portions are taken from a great many points in the pile), and in this way the probability of securing an average sample is increased. Such a sample can be secured by means of the device shown in Fig. 2 (seen clearly on the left side of the figure). In this figure A is a steam-jet aspirating pipe joined by means of a rubber tube to the filter flask B, which in turn is joined by a second tube to the pipe C, through which the gas from the stack, or pass, is being aspirated. The gas sampling bottle D is joined to the pipe C, and as the mercury is displaced by the gas it fills a second sampling bottle not shown.

A useful device for drawing a sample from a number of points in the cross-section of the stack, or pass, consists of a series of profusely perforated tubes, radiating from a small

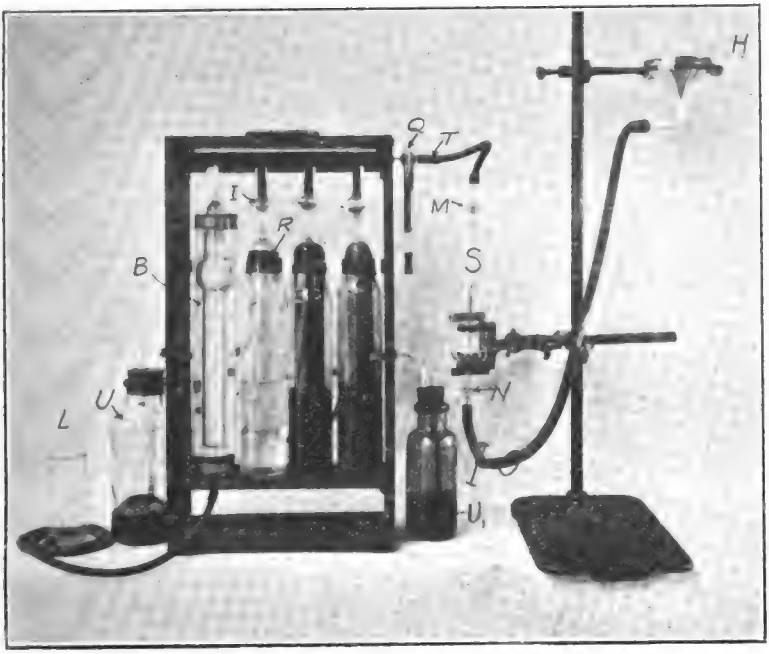


FIG. 1.

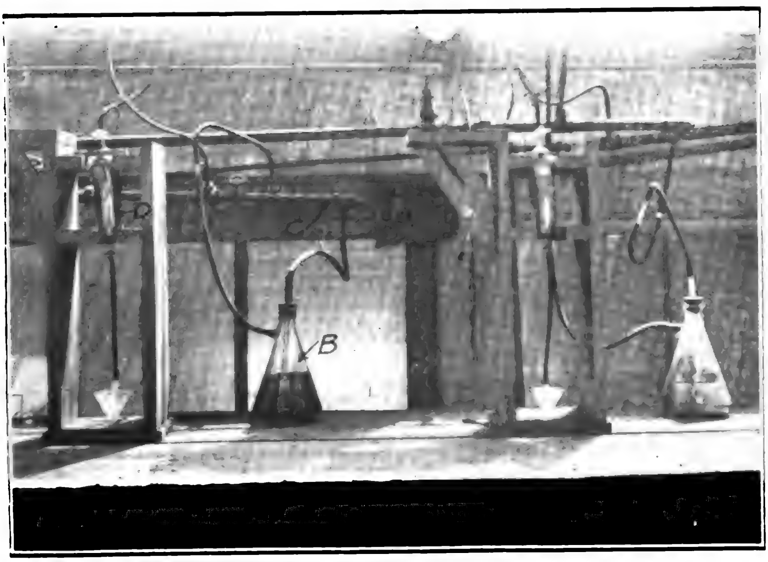


FIG. 2.

A = Steam-jet aspirator pipe.
B = Filter flask.
C = Pipe (gas) leading from stack or "pass."
D = Gas-sampling bottle.

central cylinder, closed at the bottom, but joined at the top to a tube (about $\frac{1}{8}$ inch) which passes through the side of the stack and is joined to the sampling bottle in the manner just described. A very satisfactory form of such sampling bottling is shown in Fig. 1, S.

DRAWING THE SAMPLE FROM THE STACK, OR "PASS."

The sampler having been filled with mercury and attached to the tube through which the gas is being aspirated, the mercury is allowed to flow at a predetermined rate from the lower stop cock, this rate being regulated by means of a capillary tube attached to the stop-cock, so that the time necessary to fill the sampling bottle may be made to vary from a minute or two to several hours. By interrupting the flow of mercury while there is still left a volume of five or ten cubic centimeters, an efficient seal is provided, and the sample may be kept till such a time as it may be convenient to examine it.

PREPARATION OF GAS-ANALYSIS APPARATUS FOR ACTUAL ANALYSIS.

The following directions will be more intelligible if reference is made to Fig. 1. *Filling the absorbing pipettes with the reagents.* The absorbing pipettes consist essentially of two parts: (1) a long glass cylinder constricted at the top, provided with a stop-cock, and filled with small glass tubes which serve both to subdivide the gas and to increase the absorbing surface of the reagent, and (2) a "jacket" containing the reagent into which the cylinder is immersed. To prepare the apparatus for analysis, the rubber band, Fig 1, R, which makes an air-tight connection between the jacket and the cylinder is turned down over the end of the former, which allows of its removal. Two hundred and fifty cubic centimeters of the reagent are placed in the jacket, and the cylinder, with the stop-cock open, is reinserted, the rubber band being replaced so as to make an air-tight seal between the cylinder and its jacket.

After each pipette has been filled in this manner, the next step is to adjust the level of the reagent in each to a point midway between the pipette proper and its stop-cock. To do this, first raise the leveling bottle L till the water in it flows fron T (temporarily disconnected from M), then having closed Q, lower the leveling bottle until the reagent rises to the point indicated. After all the pipettes have been adjusted, the apparatus is ready for the transfer of the sample. Fig. 1 shows the gas analysis apparatus with the sample placed ready for this transfer. After adjusting the reagents, and before connecting with the sampling bottle, the leveling bottle is raised till water flows from T. The apparatus is now ready for the sample.

TRANSFERRING THE SAMPLE TO THE GAS-ANALYSIS APPARATUS.

First, place the leveling bottle L below the level of the apparatus; attach the rubber tube T to M; open the stop-cock and allow the mercury in the capillary of this to fall into the sampling bottle. Now partially open N and allow the mercury to flow from the reservoir of mercury H, thus driving the gas into the measuring burette B. When the water in this has sunk to the 5 c.c. mark raise the leveling bottle till the water in it is level with that in the burette. Reduce the flow of the gas by partially closing N, slowly lowering the leveling bottle so as to maintain the water in it and in the burette at the same level. When the water in the latter reaches the zero mark, close the stop-cock N quickly and then Q. The measuring burette now contains exactly one hundred c.c. of gas, measured at atmospheric pressure.

DETAILS OF THE DETERMINATION OF THE CONSTITUENTS OF THE GAS.

Open stop-cock I so that its bore connects with the *front* capillary of the pipette (next to the measuring burette) which contains a solution of caustic potash (300 grams per litre) and,

lifting the leveling bottle L, pass only enough gas to clear the front capillary of the absorbing reagent; now turn the stop-cock I 180 degrees so as to connect it with the *rear* capillary of the pipette and continue to raise the leveling bottle until the water in B rises to the capillary. Again turn stop-cock I 180 degrees and lower the leveling bottle till the reagent rises to its former level. Repeat this passage of the gas twice (three passages in all). After allowing the water in the measuring burette to drain from its walls, read the level of the water in B. This gives the number of c.c. of carbon dioxide absorbed, therefore the percentage (100 c.c. of the sample having been taken). This procedure is followed in the case of the other pipettes, except that after the gas has been passed through the last pipette (which contains a solution of cuprous chloride in hydrochloric acid) the residual volume should be passed again through the first pipette (caustic potash) to absorb any hydrochloric acid vapors.

The apparatus as shown in Fig. 1 permits of the determination only of carbon dioxide, oxygen, carbon monoxide and nitrogen (by difference). The second pipette (from the left) contains an alkaline solution of pyrogallic acid (for oxygen), and the last one an acid solution of cuprous chloride (for carbon monoxide).

A larger model of this apparatus permits of the additional determinations of hydrogen, of the so-called "heavy hydrocarbons" and of methane. These latter, however, are rarely of any practical importance in flue-gas analysis.

DESCRIPTIONS AND TRIALS OF U. S. TORPEDO-BOAT DESTROYER *NICHOLSON.*

BY LIEUTENANT W. F. COCHRANE, U. S. N., MEMBER.

The *Nicholson* is one of six destroyers authorized by an Act of Congress, approved August 22, 1912. These vessels being designated as Torpedo-Boat Destroyers Nos. 51 to 56, inclusive.

The contract for building three of these boats, Nos. 51, 52 and 53, named *O'Brien*, *Nicholson* and *Winslow*, was awarded to the Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa., and was signed December 7, 1912; the time allowed for completion being 23 months for the first boat, 23½ months for the second, and 24 months for the third.

The designed speed was 29 knots, at about 1,050 tons displacement.

The contract price for each vessel was $842,000.00, of which $502,000.00 was allotted for machinery.

HULL.

Principal Dimensions.

Length between perpendiculars, feet and inches	300–0
over all, feet and inches	305–3
Beam, extreme, over plating, feet and inches	30–7
over guards, feet and inches	31–1
molded, feet and inches	30–6
on L.W.L., molded, feet and inches	30–4
Draught, mean, feet and inches	9–5
Displacement, tons	1,050
Area immersed midship section, square feet	196.6
L.W.L. plane, square feet	6,050
wetted surface, square feet	9,760
Displacement per inch at L.W.L.	14.39
Coefficient of fineness, block	0.426
midship section	0.684
L.W.L.	0.660

The general arrangement of the vessel is the same as the *Aylwin* class which were described in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XXVI, No. 1, February, 1914, except that the propellers are located farther forward, eliminating one pair of struts and shortening the shafting. Also the steering engine is located on the main deck in the after house instead of in the pilot house, where it was in the previous vessels.

MACHINERY.

The machinery is similar (except for some details) to that of the *Aylwin* class, which was fully described in the above-mentioned article. The turbines are designed to develop 17,000 shaft horsepower at about 600 revolutions per minute. There are two Cramp-Zoelly turbines arranged on two shafts with a compound, reciprocating, cruising engine on the forward end of each shaft. Between the cruising engine and the turbine there is a hydraulically operated friction clutch. This type of clutch was fully described by its designer, Mr. J. F. Metten, Chief Engineer of the Wm. Cramp & Sons S. & E. B. Co., in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XXVI, No. 1, February, 1914. An improvement in detail over the *Aylwin* type turbine consists in an improved type of nozzle rings. In the *Aylwin* turbine the nozzles were formed of thin steel guide blades in the annular openings in the cast-iron nozzle rings. The guide blades were set in the mold when the nozzle rings were cast. In many cases it was found that the guide-blade angles were not correct, and to remedy this trouble the *Nicholson's* nozzle rings are made of wrought steel with guide blades of the same material as the regular turbine blades. The blades are set in the brass container rings made in sections, and these containers are set in the annular openings of the nozzle rings and are secured by riveting over their edges into recesses provided in the nozzle rings. Conditions encountered in service by the

vessels of the *Aylwin* class have required the following difference among the auxiliaries of the *O'Brien* class vessels:

The installation of a fire and bilge pump in each fireroom; an additional distiller; an air compressor for running pneumatic tools, etc., in addition to the torpedo air compressor. The electric plant was increased from two 10-kw. generators to two 25-kw. generators; the oil-cooler circulating pump was omitted, the circulating water for the oil cooler on the *Nicholson* being taken from the main circulating-pump discharge with an emergency connection from the firemain.

Shafting and Bearings.

Cruising-engine crankshaft, length, feet and inches	5–11$\frac{3}{16}$
diameter, inches	5½
diameter of hole, inches	2
Intermediate shaft, length, feet and inches	9– 4½
diameter, inches	5¼
diameter of hole, inches	none.
Rotor shaft, length, feet and inches	18–10$\frac{5}{16}$
diameter at bearings, inches	11¼
diameter of hole, inches	7
Line shaft, forward section, length, feet and inches	10– 4½
diameter, inches	8½
diameter of hole, inches	4½
after section, length, feet and inches	20– 2¾
diameter, inches	8½
diameter of hole, inches	4½
Stern-tube shaft, length, feet and inches	25– 7¼
diameter, inches	8⅝
diameter of hole, inches	4½
Propeller shaft, length, feet and inches	22– 1½
diameter, inches	8⅝
diameter of hole, inches	4½
Rotor-shaft bearings, number of	2
diameter, inches	11¼
length, inches	16½
Cruising-engine crankshaft bearings, number of	3
diameter, inches	5½
length, forward, inches	7½
middle, inches	9$\frac{3}{16}$
after, inches	11⅞
Intermediate shaft bearings, number of	1
diameter, inches	5¼
length, inches	10

Thrust collars on shaft, number of	9
thickness, inch	¾
space between, inches	1¼
outside diameter, inches	12½
inside diameter, inches	8 13/16
bearing surface, ahead, square inches	493.8
astern, square inches	555.5
Thrust rings, number of, ahead	8
astern	9
effective thrust surface, ahead, square inches	463.0
astern, square inches	520.9
Line-shaft bearings, number of, each side	2
diameter, inches	8½
length, inches	16
Stern-tube bearings, diameter, inches	9½
length, forward, inches	27
after, inches	43
Strut bearings, diameter, inches	9½
length, inches	48 3/16
Main condensers, cooling surface, each, square feet	6,000.9
total, square feet	12,001.8
tubes, number of, each	2,934
outside diameter, inch	⅝
length between tube sheets, ft. and ins.	12–6
thickness, B.W.G.	18
Auxiliary condenser, cooling surface, square feet	306.5
tubes, number of	336
outside diameter, inch	⅝
length between tube sheets, ft. and ins.	5–6¾
thickness, B.W.G.	18
Feed-water heaters, number of	2
heating surface, each, square feet	250
total, square feet	500
tubes, number of, each	238
outside diameter, inch	⅝
length between sheets, ft. and ins.	6–5⅛
thickness, B.W.G.	18
Propellers, twin-screw, machined faces	manganese bronze.
number of blades, each	3
diameter, feet and inches	7–8½
pitch, feet and inches	6–8
pitch ÷ diameter	.865
projected area, square feet	28.21
helicoidal area, square feet	31.50
disc area, square feet	46.67
Boilers, 4 White-Forster	oil burning.
working pressure, pounds per square inch	260

Boilers — external height, feet and inches .... 12–7$\frac{7}{16}$
external length, feet and inches .... 13–2$\frac{13}{16}$
external width, feet and inches .... 14–7$\frac{3}{8}$
number of burners, each 11
heating surface, each, square feet 5,400
total, square feet 21,600
combustion space, each, cubic feet 452
heating surface ÷ combustion space 11.9
tubes, number of, each 3,256
outside diameter, inch 1
thickness, B.W.G. 12
mean exposed length, feet 6.34
Smoke pipes, number of 4
height above oil burners, mean, feet 34
area, each, square feet 16.57
combustion space ÷ area of smoke pipe 27.28
Forced-draft blowers, number of 4
type, Keith fans.
driven by Terry steam turbines.
Electric generators, number of 2
rated output, each, kilowatts 25
type, Diehl, driven by Terry steam turbines.
Evaporators, number of 2
type, Reilly multicoil.
heating surface, each, square feet 61.50
Distillers, number of 3
type, straight-tube cooling surface, each, sq. ft.... 20.54
Fuel-oil heaters, number of 2
type, U-tube.
heating surface, each, square feet 29.53
Lubricating-oil cooler, type, straight-tube cooling surface, square feet 175.22

Pumps.

Purpose.	Number.	Type.	Diameter, inches.		
			Steam cylinders.	Water or air cylinder.	Stroke, inches.
Main air	2	Blake, twin vertical beam	14	28	18
Main circulating	2	Cramp, engine-driven centrifugal.	8	—	6
Main feed	2	Blake, vertical simplex	15	10	16
Auxiliary feed	2	Blake, vertical simplex	15	10	16
Engine-room F. & B.	2	Blake, vertical simplex	7	7	12
Fireroom F. & B.	2	Blake, vertical simplex	7	7	12
Auxiliary condenser, air and circulating	1	Blake, horizontal simplex, comb.	6	8	7
Evaporator feed	1	Blake, vertical simplex	4½	6	6
Distiller fresh water	1	Blake, vertical simplex	3½	4	4
Lubricating oil	2	Blake, vertical simplex	4½	6	6
Fuel-oil service	4	Blake, vertical duplex	6	3½	8
Fuel-oil booster	2	Blake, vertical simplex	4½	6	6
Air compressor	1	Westinghouse, vertical simplex	11	11	12

TRIALS.

The contract required:

(*a*) A progressive trial over the measured-mile course at Lewes, Del., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engines connected and in use.

(*e*) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engines connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engines.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engines connected and in use, at an average uniform speed of 12 knots, as nearly as possible.

(*g*) In addition to the above-enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel oil and water consumption to be carefully measured on this trial.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the

trials, would not exceed 692.25 pounds at guaranteed maximum speed, 444.6 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

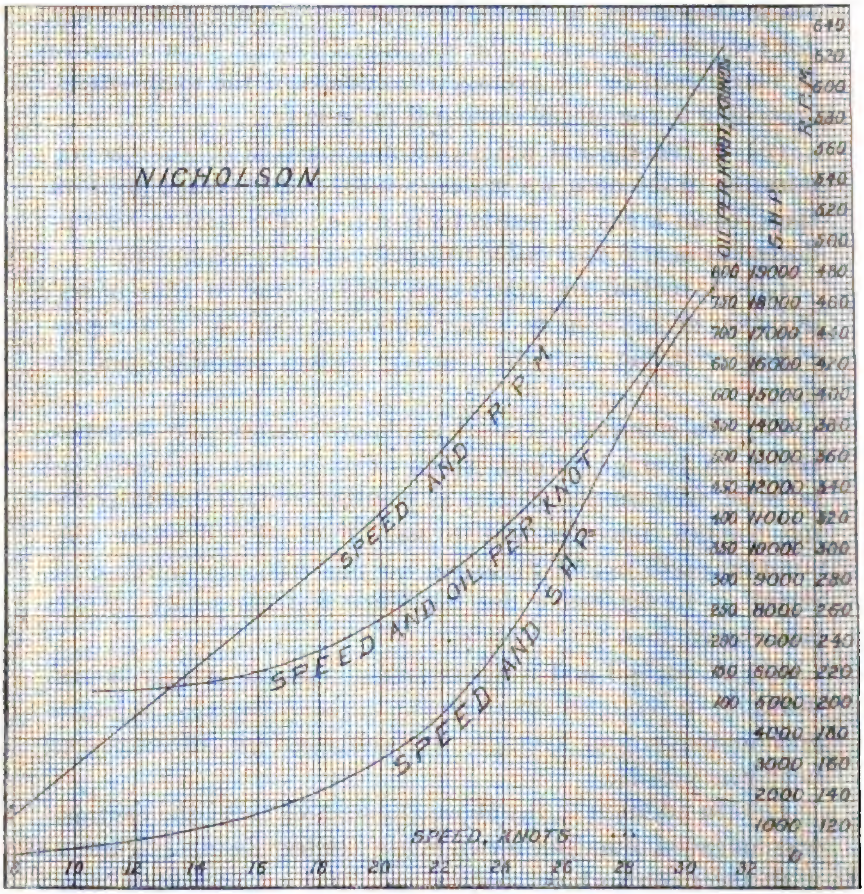


PLATE I.

Standardization Trial (a).—This trial was conducted under favorable weather conditions on the measured mile at Lewes, Delaware. Twenty-eight runs were made over the course at

various speeds, and from the data obtained the curves in Plate I were plotted. Table I gives the standardization data.

From the curve in Plate I the following r.p.m of the propellers were found to be necessary for the various speeds:

12 knots	195.00
15½ knots	252.00
24 knots	415.40
29 knots	563.40

Four-Hour Full-Speed Trial (b).—This trial began at 8:40 A. M., March 24, 1915, off Five Fathom Bank Lightship, and was completed at 12:40 the same day.

The weather was fair. The data obtained on this run is given in Table II. The machinery operated excellently. All guarantees were easily attained.

Four-Hour 24-Knot Fuel-Oil and Water-Consumption Trial (c).—Following the full-speed trial, the 24-knot trial began at 1:35 P. M., and was completed at 5:35 P. M. The weather was fair. The trial was very successful. For data, see Table II.

Two-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (g).—The trial began at 5:50 P. M., following the 24-knot run, and was completed at 7:50 P. M. The trial was very successful. Data is given in Table II.

Four-Hour 12-Knot Fuel-Oil and Water-Consumption Trial (f).—This trial commenced at 6:10 A. M., March 25, 1915, and was completed at 10:10 A. M. the same day. The weather was fair and the trial very successful. For data, see Table II.

Four-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (d).—The trial began at 10:15 A. M., March 25, 1915, and was completed at 2:15 P. M. the same day. The trial was very successful. The weather was fair. For data, see Table II.

Ten-Hour 15½-Knot Endurance Trial (e).—This trial began at 2:30 P. M., March 25, 1915, and was completed at 12:30 A. M., March 26, 1915. The weather was fair. The trial was very successful, and the reciprocating engines and clutches operated excellently.

TABLE I.—STANDARDIZATION U. S. S. "NICHOLSON," LEWES, DELAWARE, MARCH 23, 1915.

No. of Run.	Speed.	R.P.M.			S.H.P.			Speed, Average group.	S.H.P., Average group.	R.P.M. Average group.
	Knots.	*Stbd.*	*Port.*	*Mean.*	*Stbd.*	*Port.*	*Mean.*			
1	7.27	140.63	140.48	140.56	141	142	283			
2	9.82	140.43	140.96	140.70	147	150	297	8.69	299	141.44
3	7.83	143.65	143.91	143.78	165	153	318			
4	13.00	199.74	199.91	199.83	379	385	764			
5	11.64	198.72	198.30	198.51	397	371	768	12.25	758	199.09
6	12.71	199.67	199.30	199.49	369	363	732			
7	15.33	254.17	253.31	253.74	813	756	1,569			
8	15.70	252.90	253.02	252.96	758	743	1,501	15.58	1,542	253.45
9	15.58	254.01	254.23	254.12	800	798	1,598			
10	20.62	342.20	340.64	341.42	1,915	1,897	3,812			
11	20.91	340.94	341.29	341.12	1,857	1,866	3,723	20.73	3,738	341.30
12	20.48	341.97	341.06	341.52	1,863	1,830	3,693			
13	24.78	426.52	423.48	425.00	3,985	3,837	7,822			
14	23.78	420.85	419.31	420.08	3,827	3,735	7,562	24.25	7,632	421.56
15	24.66	421.62	420.55	421.09	3,813	3,768	7,581			
16	25.62	467.64	468.06	467.85	5,328	5,259	10,587			
17	26.51	468.56	471.90	470.23	5,291	5,350	10,641	26.10	10,604	469.41
18	25.75	469.20	469.37	469.29	5,345	5,203	10,548			
19	29.36	566.65	553.48	560.07	8,211	7,648	15,859			
20	28.66	569.75	559.88	564.82	8,484	7,765	16,249	28.98	16,049	562.77
21	29.22	564.01	558.74	561.38	8,088	7,749	15,837			
22	30.00	607.17	605.32	606.25	9,314	8,671	17,985			
23	30.74	610.83	606.33	608.58	9,553	8,869	18,422			
24	30.15	610.48	611.03	610.76	9,517	8,567	18,084	30.51	18,313	611.22
25	30.85	615.79	610.91	613.35	9,569	8,874	18,443			
26	30.53	619.14	617.15	618.15	9,714	8,903	18,617			
27	14.49	..........Backing.								
28	14.12									

TABLE II.—U. S. S. "NICHOLSON."

	4 hour full power.	4 hour 24 knot.	4 hour 15½ knt.	4 hour 12 knot.	2 hour 15½ knt.
Date of trial	3/24/15	3/24/15	3/25/15	3/25/15	3/24/15
Displacement	1,045	1,047.5	1,044.5	1,051.5	1,046.9
Boilers in use	4	4	2	2	2
Heating surface	21,600	21,600	10,800	10,800	10,800
Speed, in knots	29.084	24.081	15.625	12.13	15.621
R.p.m., starboard	566.40	417.35	253.91	197.15	253.85
port	565.79	417.06	253.80	196.98	253.97
mean	...	417.21	253.88	197.07	253.91
S.H.P., mean	566.10	7,257	1,624	866	1,503
I.H.P., cruising engines	15,906	...	723	375	...
Pressures:					
Main steam (G)	250.8	248.2	251.9	250.6	222.8
Main turbines:					
Full speed (abs.) S.	201.3	70.4	17.4	10.0	22.0
P.	215.4	75.0	15.0	10.0	23.2
Cruising (abs.) S.	149.9	195.5	37.0	18.0	67.6
P.	164.8	209.0	38.4	19.4	72.9
10th stage (abs.) S.	24.2	12.6	3.0	1.1	4.8
P.	26.2	12.6	3.3	1.6	4.8
14th stage (abs.) S.	14.6	7.3	.5	.56	2.4
P.	14.0	6.6	1.5	.50	2.5
Gland steam gage	6.3	4.9	3.0	2.1	4.7
Cruising engines:					
H.P. valve chest (G)	...	...	221.5	223	...
			224.6	137.8	...
L.P. receiver (G)	...	...	28.8	7.4	...
			30.0	9.3	...
Vacuum, starboard	26.6	28.3	28.3	28.5	27.6
port	27.8	28.8	29.0	29.0	28.2
Auxiliary exhaust	8.0	5.1	4.0	5.6	5.3
Forced lubrication system	12.6	8.4	10.4	9.9	10.0
Oil to clutches	...	...	75.0	75.0	...
Temperatures:					
Air	49.5	47.4	48.6	44.1	46.4
Engine room	80.0	71.3	86.3	79.1	70.1
Auxiliary room	66.8	69.5	63.3	62.5	65.3
Fireroom	71.7	71.7	85.0	84.7	77.1
Oil to cooler	104.9	95.6	70.1	64.0	85.0
Oil from cooler	84.9	75.9	41.6	41.0	66.0
Main injection	46.0	42.0	49.0	39.0	40.0
Main discharge	76.1	62.6	50.0	49.0	54.5
Fuel oil to heaters	47.0	62.5	70.0	...	...
Fuel oil from heaters	57.1	95.6	...	69.0	66.6
Smoke pipes	...	...	...	...	...
R.p.m. blowers	1,495	110.0	750.6	580	...
Air pressure	6.2	2.5	1.3	1.0	1.9
Water consumption, pounds per S.H.P., M.E.	16.224	18.632	22.06	24.22	33.94
Fuel oil, per knot run	687.3	397.74	155.95	131.86	222.42
Water evap. per pound oil	12.9	14.12	14.7	13.12	14.69
Propeller data, diameter, ins.	92½	...	...	...	...
pitch, inches	80	...	...	...	...

THE PNEUMERCATOR.

A DESCRIPTION OF ITS INSTALLATION AND OPERATION.

By Henderson B. Gregory, Associate.

GENERAL DESCRIPTION.

The Pneumercator was invented by Mr. Harry S. Parks, of Philadelphia, Pa., with intention of providing an apparatus that would measure the depth and volume of the contents of tanks, reservoirs, etc., with extreme accuracy and register this measurement at any convenient place at any distance from such tank or reservoir.

It is now manufactured in various models to meet the requirements of its application, all of which work on the same principle and consist of essentially the same elements.

These elements are—

(1) A balance chamber;

(2) A mercury gage, which is calibrated in feet and inches and the corresponding weight or volume;

(3) A pump or other means of furnishing compressed air;

(4) A control valve, directly attached to the mercury gage and also connected, through small piping, to the balance chamber and air pump.

Figure 1 shows a diagram arrangement of the component parts, and Figure 2 a detail of the control valve.

By placing the handle of the control valve in its various positions, connection is made exclusively, (*a*) between the balance chamber and the air pump, or (*b*) between the balance chamber and the gage, or (*c*) the reading of the mercury gage can be returned to zero without losing the pressure

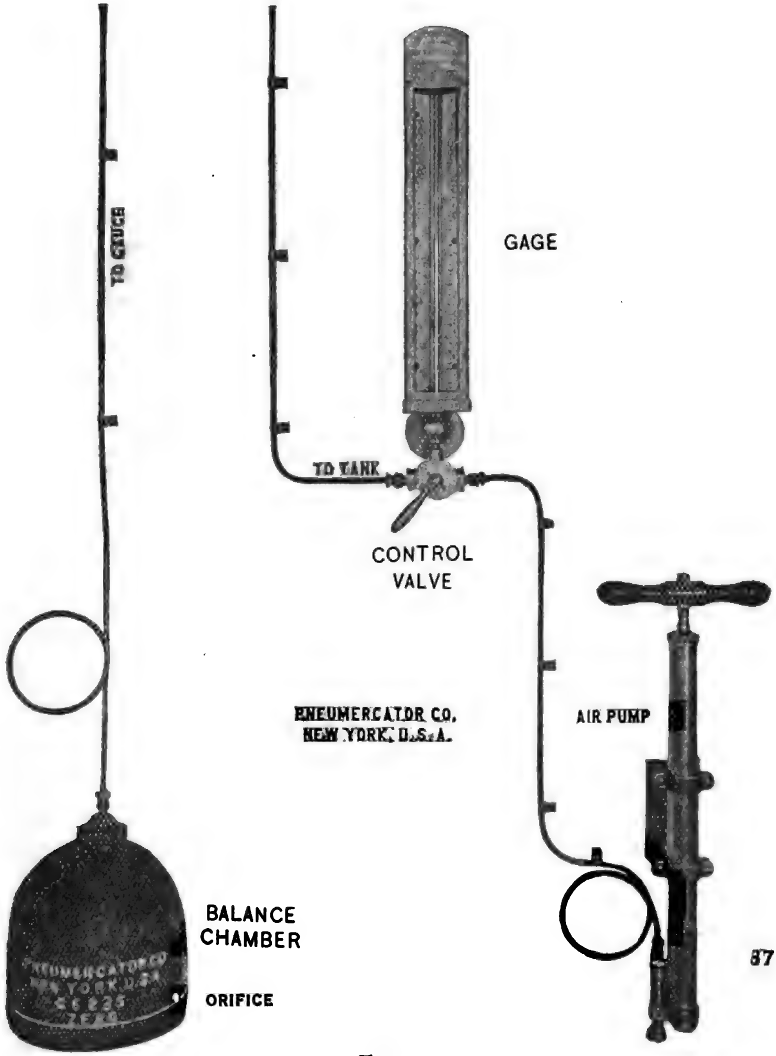


FIG. 1.

then existing between the balance chamber and the mercury gage. There is a fourth position (*d*) in which the various connections are all closed. The various positions of the valve handle, for these several functions, are defined by an index on the valve marked "GAGE"—"SHUT"—"AIR"—"VENT," corresponding to (*b*), (*d*), (*a*) and (*c*), respectively.

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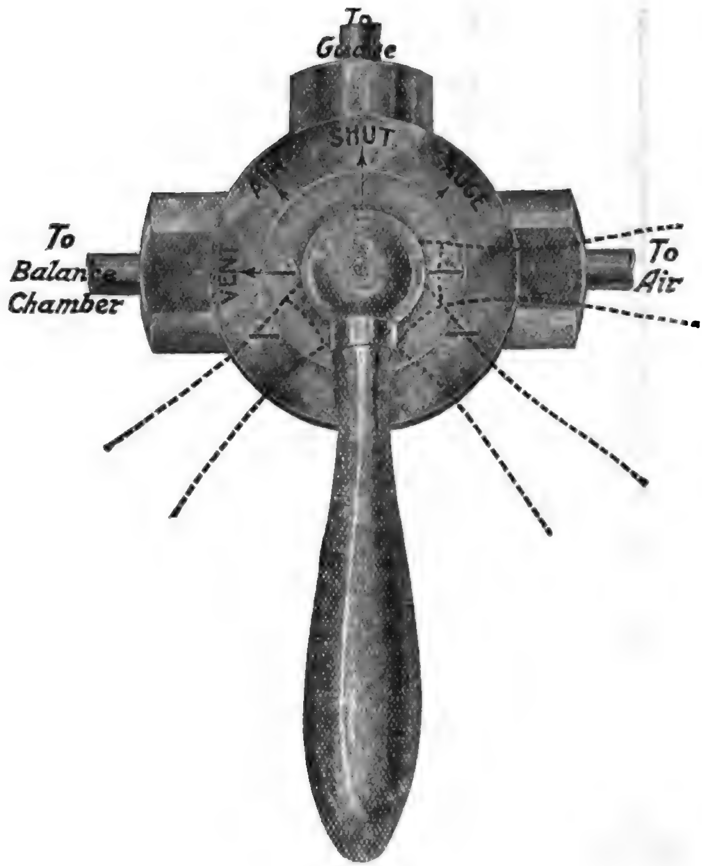


CONTROL VALVE.

FIG. 2.

A typical installation is shown in Fig. 3, a description of which follows:

The balance chamber, see Fig. 1, is placed inside the tank and securely attached so that the orifice is at some predetermined point, which point is desired to be the zero or lowest reading on the scale of the mercury gage. The pipe line is then led to the place where the reading of the contents of the tank is desired to be taken and connected to the control valve of the mercury gage. The source of

compressed air is then connected to the control valve, whereupon the instrument is ready to be placed in operation, as soon as the liquid to be measured enters the tank and rises sufficiently to cover the orifice in the balance chamber. The position of the control valve should then be at "gage." As the liquid rises in the tank it flows into the orifice in the balance chamber until its weight or head compresses the air in the pipe line (which is now sealed at the top by the mercury) so that one balances the other.

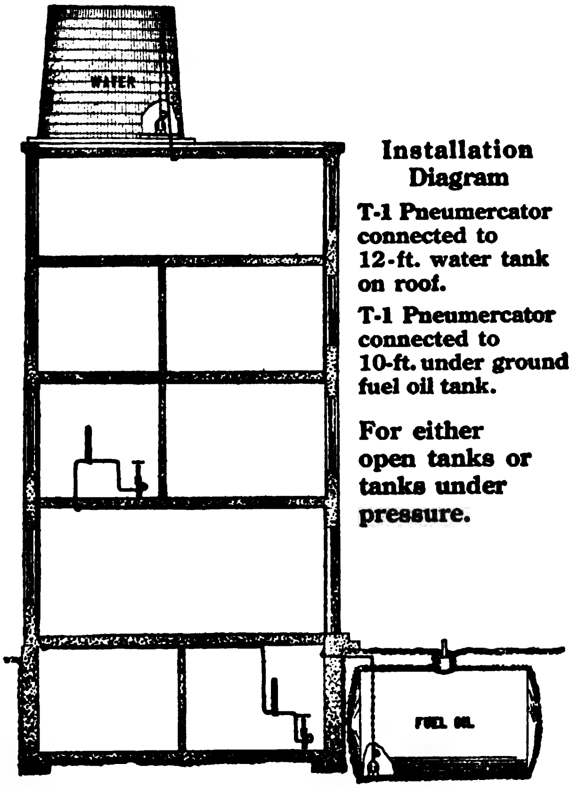


FIG. 3.

The point at which this balance takes place varies with the length of the connecting pipe line, and consequently the accuracy of the reading of the mercury gage cannot be depended

upon unless the point of balance takes place *exactly* at the orifice, as the mercury gage will show an error proportionate to the distance at which the liquid *in the balance chamber* stands above the orifice.

It will be readily seen, however, that if the level in the balance chamber where the air and liquid meet is established at the orifice, the mercury in the gage will indicate the true depth of the liquid in the tank above the orifice. Therefore, to accurately set the gage, the control valve is placed in the position marked "air," the connection then being between the balance chamber and the pump. A few strokes of the pump are sufficient to force air through the pipe line to the balance chamber at a pressure in excess of that created by the head of liquid in the tank thereby expelling whatever liquid may be in the balance chamber *above* the orifice.

The control valve is then placed in the position marked "shut," and any excess of air pressure in the pipe line greater than the head of the liquid escapes through the liquid, and the air pressure in the pipe line and balance chamber exactly balances the head of the liquid *at the orifice.*

The control valve is again placed in the position marked "gage," when the air in the pipe line exerts its pressure on the mercury so that the latter rises to a height sufficient to balance the air pressure, thus indicating the depth of the liquid in the tank above the orifice or zero point. As the tank is filled the mercury continues to rise in direct ratio to the head of the liquid.

The transverse area of the balance chamber is made very large in comparison with the bore of the mercury tube so that the volume of liquid equal to that of the mercury will occupy only an inconsiderable height in the balance chamber. Of course enough air is displaced to raise the mercury column and an equal volume of liquid enters the balance chamber to take its place, but this volume is comparatively small, and when spread out to the area of the balance chamber is of a depth so slight as to be incapable of measurement, so that no inaccuracy results therefrom.

Such displacement of air through the tubing is so slight and the motion of the mercury rising to the height which balances the pneumatic pressure is so soon accomplished that almost instantly after connection is made between the balance chamber and the indicator, the pressure becomes *static*. Thus the friction of the air in the tubes and chambers has no vitiating effect and the indicator may be located at any distance from the source of pressure.

As the indication is effected through pneumatic pressure of a gas of which the weight is negligible, the indicator may be located below the body of liquid as well as above it and will show an equal degree of accuracy regardless of the length or direction of the connecting pipe line.

The apparatus may also be used when the pressure in the tank is greater or less than atmospheric as well as under atmospheric pressure. In such cases the upper part of the tank is connected by an additional pipe line with the top of the mercury tube. This pipe line carries the pressure or vacuum as the case may be; and the usual pipe line, the pressure or vacuum plus the head of the liquid, and both the liquid to be measured and the mercury column being exposed to the same pressure, the indicator registers the head *only*. Of course should there be a leak in the pipe line between the balance chamber and the control valve or in the control valve itself, the pneumatic pressure caused by the head of the liquid would escape and the mercury gage would give a false reading, therefore it is well to re-establish the true head by means of the pump or other air supply from time to time as the tank is being filled or emptied.

In case the instrument has not been read for some time it is always advisable to "blow out" the system to see if any leak has occurred; if after this is done and the control valve returned to "gage" position the reading is the same as before, no further test is necessary, but if the second reading is less than the first, the control valve should be placed in the "vent" position; this allows the pressure on top of the mercury in the mercury cistern to escape and the mercury (then being under

atmospheric pressure) returns to zero. The control valve is then turned to " gage," and if the mercury rises to the same height as it did first, after " blowing out," these readings are the correct ones.

By these two means the instrument in itself provides a positive check of its accuracy.

INSTALLATION ON U. S. S. " NEW YORK."

The Pneumercator was first brought to the attention of the Navy Department early in the year 1913, and, following a test of its accuracy conducted at the Navy Yard, New York, the results of which aroused considerable interest and speculation, an installation was authorized on the battleship *New York,* for a practical trial aboard ship under service conditions, in connection with the fuel-oil tanks.

The type of instrument installed is shown in Fig. 4, and Fig. 5 illustrates, in diagram, the general method of connecting the apparatus to the inner-bottom tanks aboard ship.

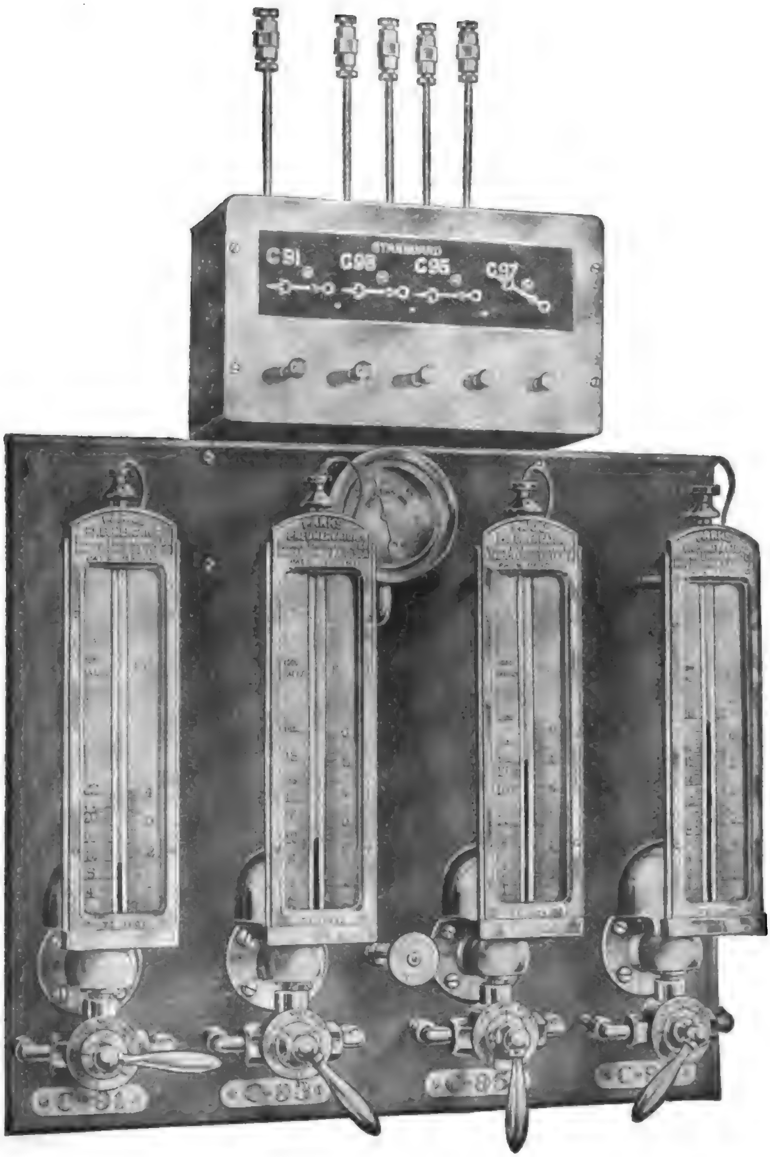
The fuel oil on the *New York,* which is intended only for auxiliary service, is carried in eight double-bottom tanks of various sizes and depths.

A balance chamber was installed in each tank, from which a ¼-inch outside diameter lead pipe was led through an oil-tight connection in the tank tops, from which point a copper pipe of the same size was carried to the engineers' office on the gun deck of the vessel.

The indicating gages were mounted on slate boards attached to the athwart-ship bulkhead.

A connection was made from the ship's compressed-air line, through a reducing valve, to two needle valves mounted on the boards, which in turn admitted the air to the control valve of each instrument and a pressure gage connected with the air line to register the air pressure available.

The scales of the mercury gage were calibrated on one side in feet, inches and half inches, and on the other in the corresponding gallons of oil at 60 degrees F., temperature. At



ONE OF THE SETS OF INSTRUMENTS ORIGINALLY INSTALLED ON U. S. S. "NEW YORK." (NOW REPLACED, EXCEPT ANNUNCIATOR, BY LATEST MODEL, FIG. 7.)

FIG. 4.

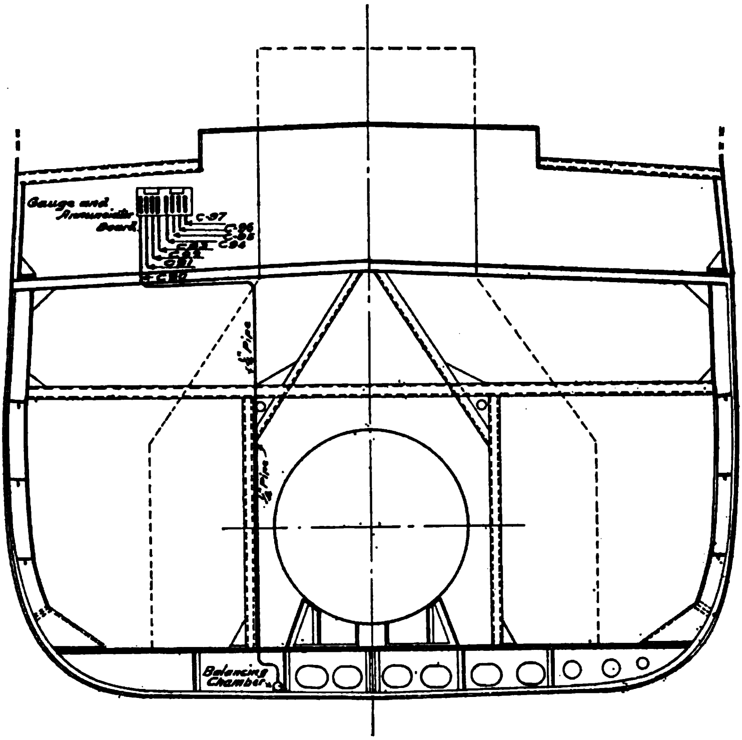


DIAGRAM ARRANGEMENT OF METHOD OF CONNECTING THE APPARATUS TO THE INNER-BOTTOM TANKS ABOARD SHIP.

FIG. 5.

the suggestion of the manufacturer an annunciator was mounted on each board which showed a signal and rang a bell when each tank was filled to the 95 per cent. full point.

To accomplish this a low voltage electric current was connected to the mercury cistern of each instrument, and a copper wire was inserted into the mercury tubes from the top reaching down to a point corresponding to the 95 per cent. full mark on the scales; consequently when the mercury rose to this point the circuit was closed and the annunciator operated.

While this method of ascertaining when the oil had reached a predetermined height was considered of the greatest value, as it did away with the necessity of all electric wiring to the tanks themselves and electrical connections to floats or

diaphragms in the tanks (the means formerly employed) and the subsequent danger of short circuits and sparks in the vicinity of the fuel oil, it was not entirely satisfactory, as the spark caused by the contact of the mercury and the copper wire in the glass tube caused the tube to become clouded and brittle.

The manufacturers, however, have now remedied this defect by having the electric contact take place in an insulated tube parallel to and back of the registering tube. This type of instrument is shown in Figs. 6 and 7, and is now installed on the battleship *New York*.

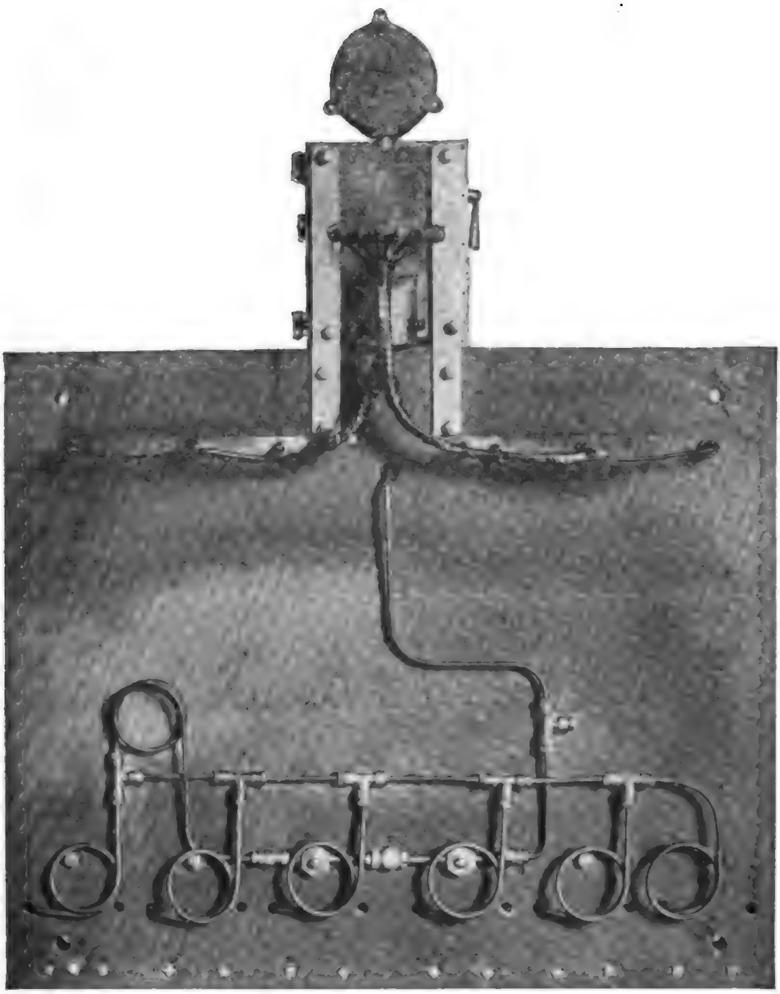
The instrument can also be arranged to operate an annunciator when the tank is emptied to some predetermined point.

The results obtained by the installation on the *New York* are most gratifying. Not only has it been possible to ascertain, with accuracy and at all times, the quantity of fuel oil in the various double-bottom tanks, but a record of the oil burned on any run has been readily obtained, and with a degree of accuracy heretofore impossible, by merely taking the difference of the gage readings at the beginning and end of the run. Reports from the vessel on the operation of the instrument pronounce it entirely satisfactory.

DRAUGHT INDICATOR FOR VESSELS.

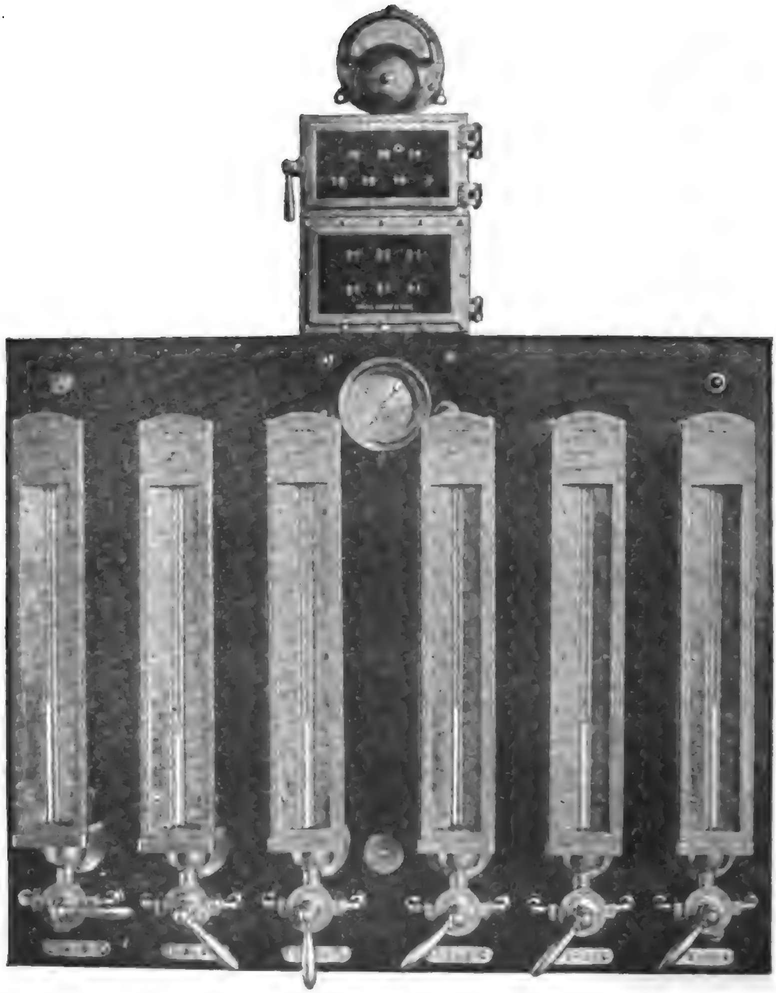
Another application of the pneumercator is that of measuring the draught of vessels. This instrument has not as yet been tested by the Navy, but the reports received from the owners of various merchant ships in which it has been installed are so interesting that a brief description of the instrument and its operation seem desirable.

In this type of instrument, shown in Fig. 8, the balance chamber is placed in a casing, which in turn is connected with the sea itself by means of a 1-inch sea valve. Two balance chambers are used, one forward and the other aft, the sea connection being installed below the light draught trim of the vessel. The ¼-inch copper tubing is then led from the balance chambers to the captain's office or the chart room and



REAR VIEW, MODEL T-I, NAVY TYPE, SHOWING ELECTRIC WIRING AT TOP NECESSARY FOR OPERATING HIGH OR LOW ALARM.

FIG. 6.



FRONT VIEW, MODEL T-I, NAVY TYPE (INSTALLED ON U. S. S. "OKLAHOMA," "PENNSYLVANIA," "ARIZONA" AND E. U. A. DREADNAUGHT "RIVADAVIA.")

FIG. 7.

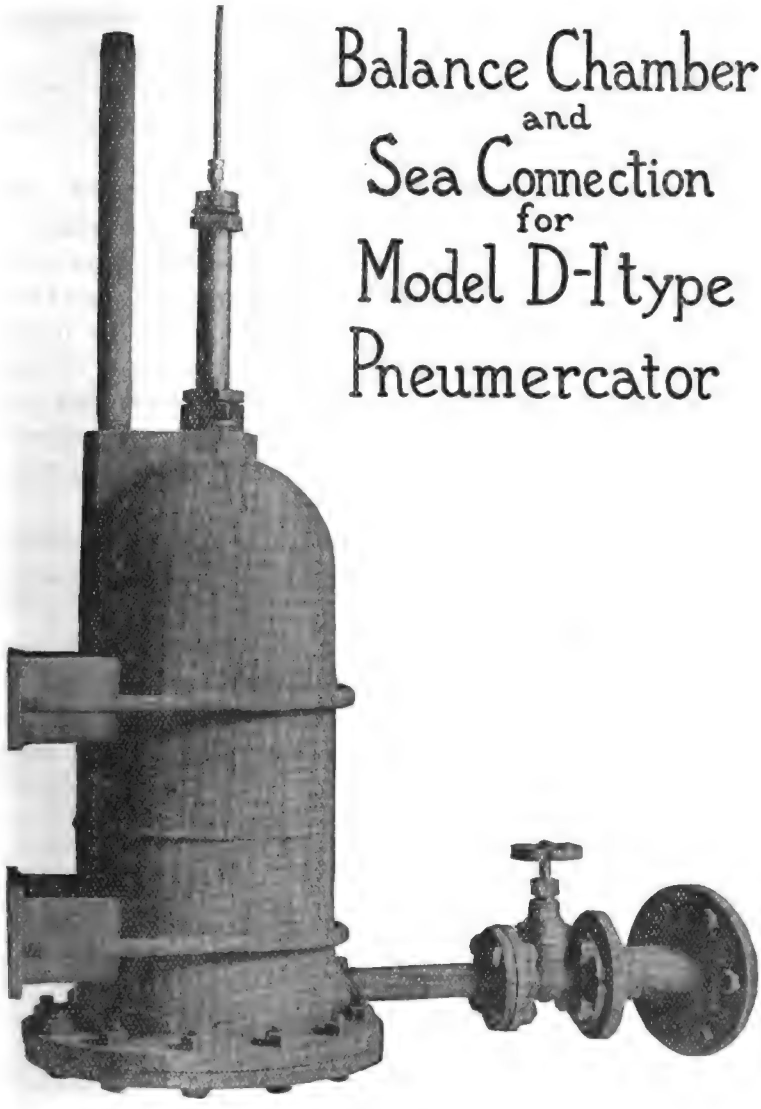


FIG. 8.

connected to the registering portion of the instrument in the same manner as in the T-I type of instrument, Fig. 7, previously described.

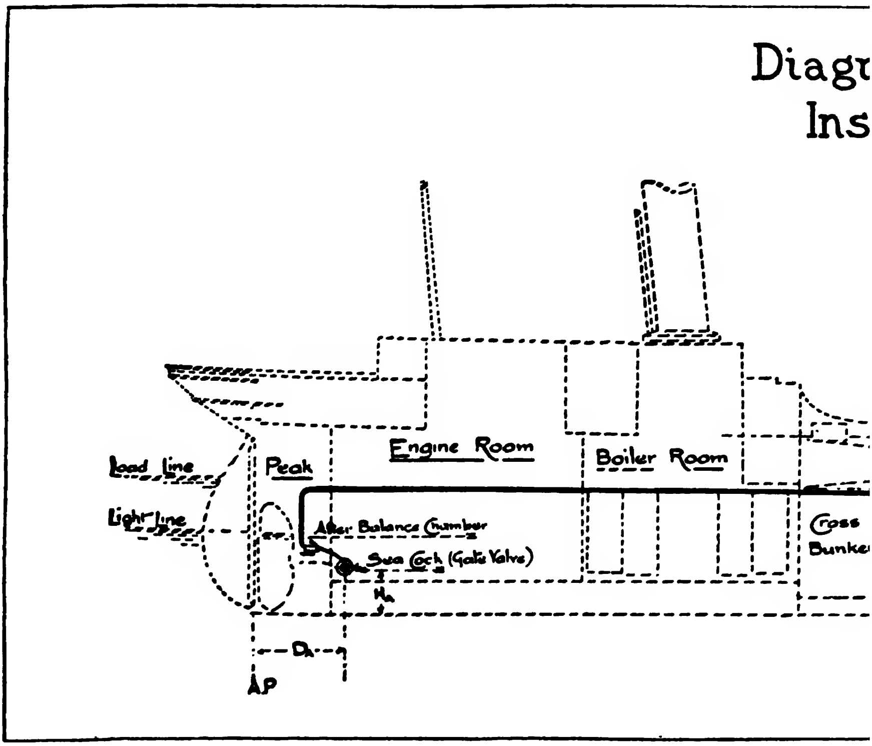
Fig. 9 shows in diagram a draught-indicator installation, and, Fig. 10, the indicating and registering instrument.

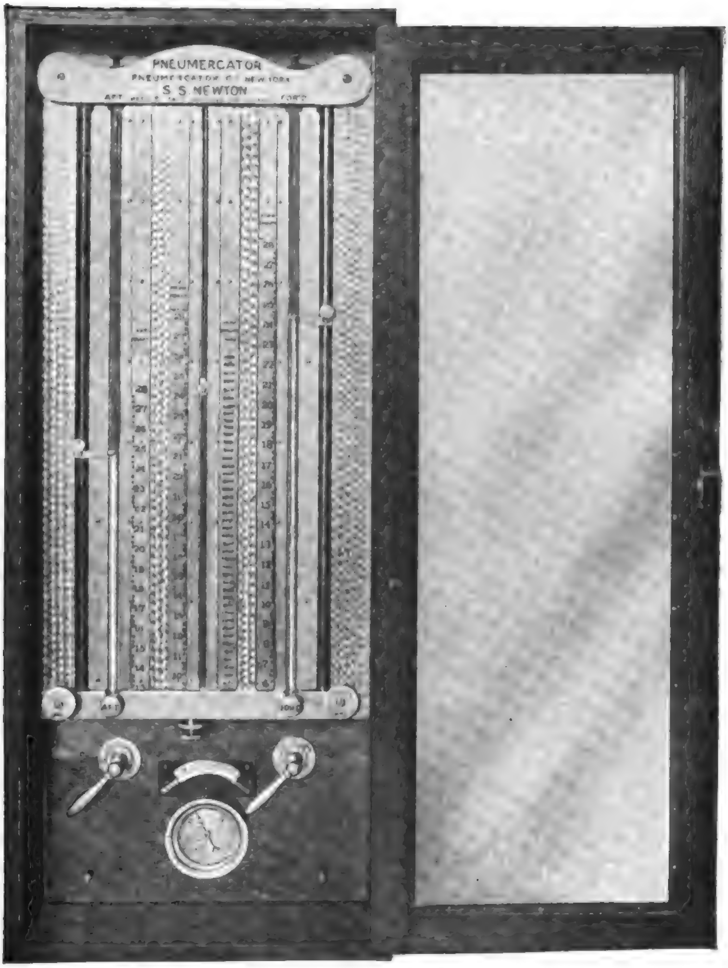
While the appearance of the balance chambers and registering instruments for the two models differs materially, their operation is practically the same.

One mercury column, Fig. 10, is marked "forward" and the other "aft," and by noting the height of the mercury in these columns and applying this to the feet-and-inches scale placed parallel to them, the draught and trim of the vessel can be noted. Equidistant from these draught scales is a mean-draught scale and a corresponding dead-weight scale. When the knife edges which travel on rods beside the mercury column are placed exactly at the top of the mercury, a central knife edge automatically registers the mean draught and the corresponding tons displaced.

If desired a third sea connection can be installed amidship and connected to its own mercury column, which will give the mean draught direct.

Furthermore, as the displacement or dead weight of a vessel represents its weight and all it contains, the amount of weight put on board or taken off a vessel can be readily ascertained by taking the difference between two successive readings of the dead-weight scale.


Diag
Ins
Engine Room
Boiler Room
Load Line
Light Line
Peak
After Balance Chamber
Sea Cock (Gate Valve)
Cross
Bunke
AP



Registering Instrument for Draught Indicator.

Fig. 10.

DESCRIPTION AND TRIALS OF THE TORPEDO-BOAT DESTROYER *DOWNES*.

By Lieutenant (J. G.) R. M. Griffin, U. S. Navy, Member.

The *Downes*, destroyer No. 45, was authorized by Act of Congress approved March 4, 1911. The other ships built under this appropriation are the destroyers *Balch*, *Benham*, *Parker*, *Aylwin*, *Cassin*, *Cummings* and *Duncan*.

The *Downes* is a twin-screw vessel fitted with a combination of Curtis turbines and reciprocating engines, and designed for a speed of 29 knots at a displacement of 1,072 tons, with main engines alone developing 16,000 shaft horsepower. She was built under contract by the New York Shipbuilding Company, Camden, N. J. The contract was signed September 8, 1911, the price being $777,500 and the time of delivery twenty-four months.

The keel was laid June 27, 1912, and on November 8, 1913, the *Downes* was launched. The preliminary official trials were held from January 5 to January 9, 1915, at Lewes, Del., and the vessel was delivered to the Government on February 11, 1915.

HULL.

The arrangement of quarters, armament and equipment is practically the same as that of the other boats of this class, which were described in the February, 1914, number of the Journal in an article on the U. S. S. *Aylwin*, *Parker* and *Benham*. The following summary gives the principal hull data:

Length over all, feet and inches	305–03
between perpendiculars, feet and inches	300–00
Breadth, molded on 9 ft. 3 ins. N.W.L., feet and inches	30–09
extreme over guards, feet and inches	31–01
Displacement for L.W.L., in tons	1,126
Tons per inch L.W.L	14.6
Load draught, feet and inches	9–11

Wetted surface, load, in square feet	9,760
Area of midship section, in square feet	189.3
Midship section coefficient	0.671
Block coefficient	0.418
Prismatic coefficient	0.622
Capacity of fuel-oil tanks, in tons	307.64
reserve feed-water compartments, tons	36.05

PROPELLING MACHINERY.

The propelling machinery consists of a combination of turbines and reciprocating engines. There are two shafts, each shaft having one Curtis turbine and a compound reciprocating engine. The reciprocating engines are designed for cruising below 15½ knots, and are coupled to the turbines by a mechanical jaw clutch, which can be thrown out of gear for higher speeds.

The cruising engines are fitted with Stephenson links, are reversible, and there is a shift valve worked by the rock shaft for shifting the exhaust from the ahead turbine to the astern turbine.

MAIN TURBINES.

The turbines are of the Curtis marine type. Each turbine was designed to develop 8,000 S.H.P. at 550 r.p.m., corresponding to a speed of 29 knots. The astern turbines are carried in the after end of the same casing as the ahead turbines and are designed to develop 50 per cent. of the ahead power.

Turbine Data.

Ahead turbine:	
1st stage nozzles:	
16 regular, throat area, square inches	10.52
3 cruising, throat area, square inches	1.2
2d stage nozzles:	
81 openings, throat area, square inches	24.22
3d stage nozzles:	
85 openings, throat area, square inches	33.74
4th stage nozzles:	
111 openings, throat area, square inches	38.183
Tip clearance buckets and casings, 5th to 36th stages, inch	0.$\frac{3}{32}$
Tip clearance nozzles and drums, 5th to 36th stages, inch	0.08
Clearance, forward side moving blades, 1st, 2d, 3d stages, inch	0.02
Clearance, forward side moving blades, remaining, inch	0.$\frac{1}{16}$
Clearance, after side moving blades, inch	0.16 to 0.28

Astern turbine:

1st stage nozzles:

10 openings, throat area, square inches	6.75
Tip clearance, buckets and casings, 2d to 10th, inch	0.$\frac{3}{32}$
Tip clearance, nozzles and drum, 2d to 10th, inch	0.08
Clearance, forward side of moving blades, inch	0.06
Clearance, after side of moving blades, inch	0.28

CRUISING ENGINES.

There are two vertical, inverted, direct-acting, compound reciprocating engines, in enclosed casings. They are fitted with Lovekin improved assistant cylinders, suspension links and reversing gear, and a shift valve to change the exhaust from the ahead to the astern turbine. The links and shift valve may be worked by hand or by steam.

Stroke, inches	14
Length of connecting rod, inches	35
Diameter of piston rods, inches	3
Cylinder, diameter, inches	H.P., 12½; L.P., 26¼

	H.P.		L.P.	
	Top.	Bottom.	Top.	Bottom.
Piston clearance, inch	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{3}{16}$
Travel of valve, inches	4½		4½	
Number and size of valves inches	One, 8½		One, 11	
Kind of valve	Piston, single port.		Piston, double port.	
Valve takes steam	Inside.		Outside.	
Depth of ports in liners, inches	$1\frac{3}{8}$	$1\frac{3}{8}$	$1\frac{7}{64}$	$1\frac{7}{64}$
Steam lap, inches	$1\frac{5}{32}$	$\frac{61}{64}$	$1\frac{9}{64}$	1
Exhaust lap, inch	$-\frac{3}{16}$	$+\frac{1}{4}$	$-\frac{13}{32}$	$+\frac{1}{32}$
Angular advance	38°-40′	38°-40′	44°-15′	
Steam lead, angular	8°	13°-15′	13°-30′	17°-30′
linear, inch	$\frac{1}{4}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{9}{16}$
Cut off, in inches	10.08	9.52	9.52	8.68
p.c. of stroke	72	68	68	62
Mean cut off p.c. of stroke	70		65	
Comp'sion, in inches	1.4	1.68	1.4	1.68
p.c. of stroke	10	12	10	12
Release, in inches	1.59	1.28	2.45	2.24
p.c. of stroke	$11\frac{3}{8}$	$9\frac{1}{8}$	$17\frac{1}{2}$	16
Steam opening, inches	$1\frac{3}{32}$	$1\frac{19}{64}$	$(1\frac{7}{64} + 1\frac{7}{64})$	
Exhaust opening, inches	$1\frac{3}{8}$	$1\frac{3}{8}$	$2\frac{7}{32}$	$2\frac{7}{32}$
Steam velocity, liner	4,070	3,260	5,410	5,340
cyl. port	3,830	3,610	6,290	6,210
Exhaust velocity through liner	3,280	3,120	5,390	5,340
R.P.M. (designed)		254		

SHAFTING AND BEARINGS.

The following table gives the data for one shaft with the sections in their proper order from forward:

Name of Section.	Shafting.			Bearings.			
	Length.	Diameter.	Axial Hole.	Number.	Diameter.	Length.	Type.
	ft. & ins.	*ins.*	*ins.*		*ins.*	*ft. & ins.*	
Cruising engine, crank shaft	8–11¼	08	3¾	4	08	0–09¼	Babbitted
Clutch coupling	18⅛	08	4	None			
Turbine shaft	22–00¾	16¼	9	2	13	0–18	Babbitted
Line shaft, for'd section	11–04	10	7	None			
Line shaft, after section.	23–06⅛	10	7	2	10⅜	0–14	Babbitted
Stern-tube shaft	23–11 5/16	10	7	1	10 29/32	2–02	Lignumvitae
				1	10 29/32	3–08	Lignumvitae
Propeller shaft	28–10 3/16	10	7	1	10 29/32	2–07	Lignumvitae
				1	10 29/32	3–08	Lignumvitae

The shaft horsepower is obtained by a Gary-Cummings torsion-meter fitted in the first section of the line shafting.

Thrust bearings of the following characteristics are provided at the forward end of each turbine:

Collars on shaft, number	7
thickness, inches	1⅛
space between, inches	3
outside diameter, inches	14⅝
inside diameter, inches	8¾
bearing surface, 6 collars, square inches	626.24
Thrust shoes, number	6
effective (thrust) surface, square inches (6)	388.44

PROPELLERS.

The propellers are three-bladed and turn outboard when going ahead. They are cast solid of manganese bronze, machined to a true helical driving surface, balanced and polished. The following are the propeller measurements:

	Starboard.	Port.
Diameter, inches	$92\frac{1}{2}$	$92\frac{1}{2}$
Pitch, inches	$81\frac{7}{8}$	$81\frac{7}{8}$
Area, helical, square inches	4,461.	4,449.
projected, square inches	4,005.	3,984.
disc, square inches	6,719.	6,719.
Ratio pitch to diameter, P/D	0.8851	0.8851
Height lower tip blade above keel, inches	3.662	
Immersion of upper tip blade at load draft light, inches	26.46	
Edge of propeller from C. L. ship, feet	2.	

MAIN AND AUGMENTER CONDENSERS.

There are two main condensers, each having an augmenter condenser. They are of oval section, with horizontally-curved tubes expanded into the tube sheets.

Data One Main Condenser.

Cooling surface, square feet	5,835
Tubes, length as fitted, feet and inches	15-4 to $15-9\frac{1}{2}$
number	2,378
thickness, B.W.G	18
outside diameter, inch	$0\frac{5}{8}$
Connections:	
Main exhaust, rectangular, square feet	15
Auxiliary exhaust, diameter, inches	7
Bleeder (one condenser only), diameter, inches	$2\frac{1}{2}$
Safety valve, diameter, inches	2
Air pump, wet suction, inches	$8\frac{1}{2}$
Air pump, dry suction (from augmenter), inches	9
Circulating water:	
Injection and discharge, diameter, inches	22

The augmenter condensers are of the Parsons type. Each has a cooling surface of 152 square feet.

MAIN AIR PUMPS AND CONNECTIONS.

There are two Blake, double, vertical, bucket, single-acting main-air pumps, located in the auxiliary room, amidships. Each pump has a 14-inch steam cylinder, two 28-inch water cylinders, 18-inch stroke, and a 12-inch suction from its own condenser, through a water seal. The water seals ate cross connected. A make-up feed line from the reserve-feed tanks

joins each pump suction. The turbine drains are led to each pump. The pumps discharge separately through 10-inch discharge pipes to the first compartment of the feed and filter tank.

MAIN CIRCULATING PUMP AND ENGINE.

There are two main circulating pumps of the centrifugal type, each driven by a reciprocating engine, with steam cylinder 9-inch diameter by 8-inch stroke.

The engine is enclosed in a casing and connected to the forced-lubrication system, and has besides a small pump of its own driven from its shaft and supplying oil from a well.

FEED AND FILTER TANK.

The feed and filter tank, of about 800 gallons capacity, is located in the forward part of the auxiliary room amidships, over the main air pumps. It is divided into two parts by a horizontal partition. The upper part, of 300-gallons capacity, is further subdivided to furnish two filter chambers. The filter compartments may be by-passed.

MAIN FEED PUMPS.

Two Blake, vertical piston, double-acting, single main feed pumps are located in the port side of the engine room forward. Each pump has a 15-inch steam cylinder, 10-inch water cylinder, and 16-inch stroke. They have 5½-inch suctions from the feed and filter tanks, 3-inch suctions from the reserve-feed tanks, and 5½-inch suctions to water seals cross connected. They discharge through 4-inch feed line to the boilers.

AUXILIARY CONDENSER AND PUMPS.

An auxiliary condenser of 250 square feet cooling surface is located amidships in the after part of the auxiliary room. The shell is cylindrical, the tubes being curved in the vertical plane and expanded into the tube sheets. The tubes are supported at the middle and are protected from the auxiliary

exhaust by a perforated baffle plate. A Blake simplex, combined air and circulating pump, 6 inches × 8 inches × 8 inches × 7 inches, is provided.

FORCED-LUBRICATION SYSTEM.

A complete system of forced lubrication is provided for the main and thrust bearings and cruising engines.

BOILERS.

The boilers are of the Thornycroft type, arranged in pairs in two separate firerooms. They are equipped for fuel oil and each is fitted with seven Schutte-Koerting burners.

Data for One Boiler.

Designed working pressure, pounds	265
Test pressure, water, pounds	400
Heating surface, square feet	6,614
Combustion chamber, length, feet	9.9
width, feet	10.989
area, square feet	108.79
volume, cubic feet	622.175
Floor to top of steam drum, feet and inches	13–3$\frac{19}{32}$
Length, external, over steam drum, feet and inches	14–2$\frac{21}{32}$
Width, external, over casing, feet and inches	16–4$\frac{1}{2}$
Area, through cone opening, square feet	13.11
through flame opening, square feet	5.7
through uptake, square feet	19.4
smoke pipe, square feet	18.0
Diameter, steam drum, inches	42
water drums, inches	19
Tubes, number	176 and 2374
diameter, outside, inches	1$\frac{3}{8}$ and 1$\frac{1}{8}$
thickness, B.W.G	11 and 12

FUEL-OIL SYSTEM.

Fuel oil is carried in tanks outside the machinery spaces. There is a separate suction line from each of the forward tanks to a manifold in the forward fireroom. Similarly separate suction lines from each of the after tanks lead to a manifold in the auxiliary room. These two manifolds are connected

by a common suction main on which either the service or booster pumps can take suction.

Two booster pumps are provided, one in each fireroom. They have suctions from all tanks and deck connections, and discharges to the service pumps, suctions, storage tanks and deck-hose connection.

There are four service pumps, two in each fireroom. They have suctions from storage tanks, direct or via booster-pump charge, and discharge to the burners.

For raising steam when no pressure is available a hand pump, capable of supplying oil at 200 pounds pressure to two burners, is fitted in each boiler compartment.

There are eleven 2.5-mm. tip burners and tuyeres of the Schutte-Koerting type. Oil is supplied to each burner through $\frac{3}{8}$-inch lines at about 200 pounds pressure. The tuyeres are so arranged that the quantity of air through each may be independently regulated at each burner.

FORCED-DRAFT BLOWERS.

In each fireroom there are two Sturtevant, vertical, single-inlet cone fans, driven by Terry steam turbines. The fan and turbine are both on the same shaft, and are supported beneath the deck. Air is drawn down through the fireroom ventilators and driven out horizontally into the firerooms. The blowers are rated at 1,600 r.p.m., at which speed they deliver 33,000 cubic feet of air at 5 inches water pressure. The two after blowers have ducts leading to the engine room, so that warm air may be drawn from it.

FEED-WATER HEATERS.

Two Schutte-Koerting spirally-corrugated film feed-water heaters are located one in each fireroom. They are of the horizontal-pressure type and use auxiliary-exhaust steam as the heating agent. Both the main and auxiliary-feed pumps can discharge through them.

Data for One Feed-Water Heater.

Total heating surface, square feet		191.1
Outer tubes,	number	32
	thickness, B.W.G	13
	outside diameter, inches	2.565
	length as fitted, feet and inches	3–10
Inner tubes,	number	32
	thickness B.W.G	14
	outer diameter, inches	2.166
	length as fitted, feet and inches	4–5½
Shell diameter, inches		27⅝
Length over all, feet and inches		6–10⅝

MAIN STEAM PIPING.

Each boiler has a 7-inch main stop valve and steam line. The lines from boilers 1 and 2 unite in the forward fireroom in a 9½-inch line which passes along the port side to the engine room. The lines from 3 and 4 boilers unite in a 9½-inch line and pass along the starboard side to the engine room. Each line has a balanced slip joint in its fireroom and a bulkhead stop in the engine room. In the engine room there are 9½-inch throttle valves to ahead and astern turbines, 3½-inch throttle valves to the reciprocating engines, 2½-inch bleeder to port condenser, and steam connection to reversing engine. There is a 6½-inch cross connection, with slip joint and stop valve. Boiler and bulkhead stops are fitted with deck operating gear.

AUXILIARY STEAM PIPING.

There is a 3½-inch auxiliary stop valve on each boiler, which takes off from the main stop casting, and is fitted with deck-operating gear. In the forward fireroom branch lines run to the auxiliaries of that room, to forward deck machinery, heating system and galley, tank-steaming connections and whistle. The 3½-inch connections from the boilers of this room unite on the starboard side in a 4-inch line which runs aft into the after fireroom. Here there is a cut-out valve, and the line is joined by a 3½-inch line with cut-out valves from the two

boilers of this room. The lines of these two boilers in addition unite on the port side and continue aft in a 4-inch line, reducing to 3½ inches after passing the main circulating engines. The line on the starboard side reduces to 3½ inches at the same point, and continues aft to the auxiliary room, where it crosses over and joins the port line. There are two cutout valves in this room, one on each side of the dynamos. The auxiliary steam line thus forms a loop embracing the after fireroom and the engine rooms, with a branch to the forward room.

AUXILIARY EXHAUST PIPING.

The auxiliary exhaust line, beginning on the port side of the forward fireroom as a 7-inch line, receives there the exhaust from all forward machinery. It passes aft on the port side, increasing to 8 inches as it receives the exhaust from the auxiliaries in the after fireroom. As it enters the engine room, it increases to 9 inches, and continues aft to the port after corner of the engine room. Here it divides into two 7-inch branches to each main condenser, and a 7-inch line which passes a spring-loaded valve to the auxiliary condenser. Steam from engine-room auxiliaries and auxiliary room, except dynamos and auxiliary condenser, unite in branches which join the 7-inch branch of the auxiliary exhaust to the starboard condenser.

In addition to the 7-inch exhaust connections to the main and auxiliary condensers, the exhaust line has 6½-inch connections to the main turbines at the 1st, 3d and 18th stages; 7-inch connections to the feed-water heaters, and 7½-inch escape pipe to atmosphere.

MAIN AND AUXILIARY FEED SYSTEM.

An 8-inch suction line runs from the feed tank with 5½-inch branches to the main and auxiliary feed pumps. The main feed pumps discharge through 4-inch discharges to a 6-inch main feed line which has 4-inch branches to each fireroom. These are so arranged that the feed water may pass by or

through the feed-water heaters, and enter the upper drum of the boilers. The auxiliary feed pumps can also discharge through or by the feed-water heaters.

The main feed pumps have in addition a 5½-inch suction to the water-seal cross-connection pipe, and 3-inch suctions to the reserve-feed tanks. The auxiliary feed pumps have 3-inch suctions to the reserve-feed tanks and 1½-inch hose connections for suction and discharge. Make-up connections join the main air-pump suctions. Overflow from the feed tanks may flow to bilge or return to the reserve tanks through a 4-inch line.

EVAPORATING AND DISTILLING PLANT.

There are two Schutte-Koerting single-effect evaporators in the auxiliary room, and above them, in the hatch, two distillers. The evaporators have a combined capacity of 3,750 gallons of potable water per 24 hours, and the distillers 2,500 gallons.

Evaporators take steam from the auxiliary steam line through 1½-inch lines and drain through a trap to the feed tank or to the starboard main condenser.

Evaporator feed is supplied by a 4½-inch × 6-inch × 6-inch Blake pump, which draws from the sea or from the overboard discharge of the distillers. It discharges through or by-passes a Department-type feed-water heater which uses the vapor from the evaporators as the heating agent. As steam forms in the evaporators it rises through 4-inch valves to a common 5½-inch vapor line to the distillers, from which it drains as water to a 50-gallon reservoir tank. A 3½-inch × 4-inch × 4-inch Blake pump pumps from this tank to the feed tank, fresh-water tanks or cofferdam. Vapor may also be admitted to the auxiliary exhaust line. Circulating water for the distillers is supplied by either fire and bilge pump.

Data for One Evaporator.

Type	Schutte-Koerting.
Diameter inside, inches	22
Height over all, feet and inches	5–9½
Rows of tubes, number	4

Tubes "U," number	32
diameter inside, inch	1
diameter outside, inches	$1\frac{1}{8}$
Water chamber	Bronze.
Heating surface, total, square feet	34.16

Data for One Distiller.

Type	Vertical, straight-tube, S. & K.
Diameter, inside, inches	$11\frac{3}{4}$
Height over all, feet and inches	$4\text{–}00\frac{1}{2}$
Tubes, number	81
outside diameter, inch	$00\frac{5}{8}$
thickness, B.W.G., number	16
length as fitted, feet and inches	$2\text{–}06\frac{5}{8}$
Cooling surface, total, square feet	32

Data for Evaporator Feed Heater.

Type	"U" tube, Bureau.
Tubes, number	24
inside diameter, inches	$1\frac{1}{4}$
thickness, B.W.G., number	14
Heating surface, square feet	18.76

ELECTRICAL PLANT.

There are two 10-kw. dynamos and switchboard on the port side of the auxiliary room. The generators, made by the General Electric Company, are driven by single-stage, three-row, Curtis turbines at 5,000 r.p.m., and deliver current at 125 volts. Lighting circuits from the switchboard lead to the machinery spaces and to panel boxes forward and aft in the living compartments. These boxes contain separate switches for each branch. Two power circuits with rheostats supply two 24-inch General Electric searchlights, one on the after deck house and the other above the bridge. A third power circuit carries current to the lathe. Call bells and the fuel-oil-filling alarm system are on a battery circuit.

There is a 2-kw. radio set, manufactured by the Garward Electrical Company. The motor generator transforms 125 volts D. C. to 110 A. C. at 550 cycles and makes 1,875 r.p.m.

MACHINERY WEIGHTS.

The following table gives the actual machinery weights:

Group.		Weight in pounds.
I. Main engines:		
Reciprocating engines, cylinders	12,825	
Turbine casings, rotors, etc	137,114 =	149,939
II. Shafting (line and propeller):		
Reciprocating engines	5,870	
Turbine engines	35,803 =	41,673
III. Main engine framing and bearing:		
Reciprocating engines	11,336	
Turbine engines	10,062 =	21,398
IV. Reciprocating parts (reciprocating engines)		3,496
V. Main engine valve gear (reciprocating engines)		3,842
VI. Main condensers, including augmenters		52,613
VII. Main air and circulating pumps		27,519
VIII. Propellers		8,753
IX. Boilers		184,676
X. Boiler fittings		47,586
XI. Smoke pipes and uptakes, including guys		17,024
XII. Steam and exhaust pipes and valves		54,136
XIII. Suction and discharge pipes and valves		38,981
XIV. Lagging and clothing		9,454
XV. Flooring, grating, ladders, etc		11,862
XVI. Auxiliaries		23,096
XVII. Fittings and gear		16,997
XVIII. Water		75,127
XIX. Stores, tools and spare parts (exclusive of parts to navy yards, and parts of electrical plant)		11,695
XX. Miscellaneous machinery, etc		60,582
XXI. Electrical plant (exclusive of conduit and wiring, but including 404 pounds for its spare parts)		3,157
XXII. Connections under steam engineering to other miscellaneous machinery		3,418
Total weight in pounds		867,024
Total weight in tons		387.06
Contract weight in tons		387.45

TRIALS.

The contract required:

(*a*) A progressive trial over the measured-mile course at Delaware Breakwater, for standardizing the screws, extending from maximum speed down to a speed of 12 knots.

(*b*) A full-speed trial of four hours' duration in the open

sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engine connected and in use.

TABLE I - STANDARDIZATION TRIAL DATA - JAN. 5, 1915 - U.S.S. DOWNES.

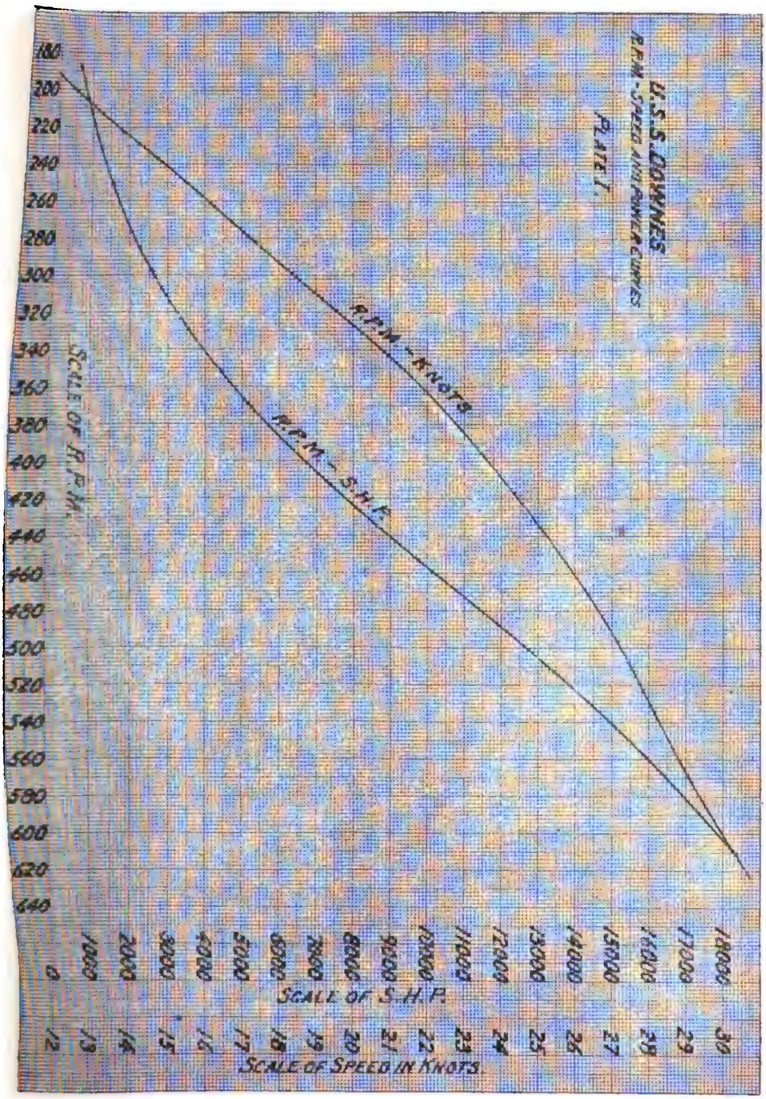
No. of Run	Time on Course.		Speed in Knots.	Revs. to go One Knot.			R.P.M.			S.H.P.		
	Mins.	Secs.		Star. Engine.	Port Engine.	Mean	Star. Engine.	Port Engine.	Mean.	Star. Engine.	Port Engine.	Total.
1	4	17.6	13.98	820.4	831.1	825.8	191.09	193.57	192.23	352	327	679
2	5	50.9	10.26	1157.9	1152.8	1155.4	198.04	197.17	197.61	392	360	752
3	4	18.5	13.93	823.4	823.5	823.5	191.07	191.09	191.08	352	336	688
Mean of Group			12.11						194.66			710
4	4	18.2	13.94	1091.9	1085.2	1088.6	253.78	252.23	253.01	849	767	1616
5	3	30.0	17.14	887.4	879.2	883.3	253.48	251.14	252.31	848	730	1578
6	4	17.9	13.96	1059.3	1050.0	1054.7	246.44	244.28	246.36	757	660	1417
Mean of Group			15.55						250.75			1547
7	3	6.9	19.32	923.7	919.5	921.6	297.55	296.22	296.89	1341	1241	2582
8	3	29.7	17.17	1042.1	1038.2	1040.2	298.31	297.18	297.75	1324	1225	2549
9	3	6.2	19.33	931.6	931.4	931.5	300.12	300.05	300.09	1373	1339	2712
Mean of Group			18.25						298.12			2598
10	2	55.9	20.47	1030.8	1032.4	1031.6	351.60	352.15	351.88	2280	2190	4470
11	2	41.3	22.32	947.0	947.5	947.3	352.05	352.22	352.14	2259	2166	4425
12	2	55.2	20.55	1028.1	1028.9	1028.5	352.07	352.35	352.21	2211	2120	4331
Mean of Group			21.42						352.09			4413
13	2	26.3	24.61	994.3	995.7	995.0	408.59	408.19	408.39	3598	3457	7055
14	2	31.4	23.78	1050.5	1057.8	1054.2	416.32	419.19	417.76	3865	3768	7633
15	2	25.9	24.67	1010.7	1014.3	1012.5	415.48	416.96	416.22	3858	3805	7663
Mean of Group			24.21						415.16			7496
16	2	17.6	26.16	1149.1	1148.9	1149.0	501.25	501.13	501.19	6399	6266	12665
17	2	6.1	28.55	1047.5	1051.5	1049.5	498.41	500.31	499.36	6431	6256	12687
18	2	19.9	25.73	1169.1	1169.8	1169.5	501.38	501.69	501.54	6504	6273	12777
Mean of Group			27.25						500.36			12709
20	2	12.3	27.21	1255.2	1256.3	1255.8	569.05	569.55	569.30	8353	8046	16399
21	1	56.5	30.90	1101.2	1103.7	1102.5	567.35	568.67	568.01	8289	8149	16438
22	2	12.7	27.13	1263.5	1265.4	1264.5	571.07	571.91	571.49	8304	8156	16460
Mean of Group			29.04						569.20			16434
23	1	53.3	31.77	1115.8	1122.2	1119.0	590.35	593.76	592.06	8826	8468	17294
24	2	10.4	27.61	1288.6	1294.5	1291.6	592.91	595.62	594.27	8305	8696	17601
25	1	52.8	31.91				593.53	597.49	595.51	8955	8723	17678
26	2	8.8	27.95	1319.6	1325.5	1322.6	596.86	599.53	598.20	8924	8793	17717
27	1	53.2	31.80	1074.3	1081.3	1077.8	595.17	599.04	597.11	8898	8786	17684
Mean of Group			29.815						595.65			17622

(*e*) An endurance trial of 20 hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engine connected and in use. Fuel-oil and water-consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engine.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engine connected

and in use, at an average uniform speed of 12 knots as nearly as possible.

(*g*) In addition to the above-enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel-oil and water-consumption to be carefully measured on this trial.



Fuel-oil Consumption Guaranties.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 700 pounds at guaranteed maximum speed, 460 pounds at 24 knots, 210 pounds at 15½ knots, and 175 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a

TABLE II – TRIAL DATA – U.S.S. DOWNES.

	4 Hrs., Full Speed Trial, (b)	4 Hrs., 24 Knot Trial, (c)	4 Hrs., 15½ Knot Trial, (d)	20 Hrs., 15½ Knot Trial, (e)	4 Hrs., 12 Knot Trial, (f)	2 Hrs., 15½ Knot Trial, (g)
Date of Trial	Jan. 6, 1915	Jan. 6, 1915.	Jan. 7, 1915	Jan. 8 & 9, 1915	Jan. 7, 1915	Jan. 7, 1915.
Speed in Knots	29.071	24.093	15.573	Endurance Test of Machinery. No data taken.	12.157	15.658
Draught, mean on trial, feet and inches	9-9⅝	9-9⅝	9-9⅝		9-10	9-10⅛
Displacement, corresponding, tons	1106	1106	1104		1112	1105
Slip of Propellers, percent, Starb'd.	24.14	13.1	8.53		7.07	8.28
Port	24.39	13.23	7.27		7.92	7.66
Mean	24.27	13.17	7.9		7.5	7.97
Engines in operation	Main Turbines	Main Turbines	Main Turbines & Cruising Eng.		Main Turbines & Cruising Eng.	M. Turb.
Number of Boilers used	4	3	1		1	1
Burners used, (11 per Boiler)	42	24⅛	5⅞		4⅛	8⅛
Heating Surface used, sq. ft.	26,456	19,842	6614		6614	6614
Sq. ft. per S.H.P.	1.622	2.769	4.329		9.872	4.292
Cooling Surface, sq. ft., (Main Cond.), per S.H.P.	0.716	1.629	7.637		17.42	7.573
Pressures, (average):						
Main Steam at Boilers, pounds gauge	251.01	246.46	245.63		240.0	241.88
Eng. Room, " "	241.75	238.25	245.63		239.63	242.62
Ahead Steam Chest, pounds absolute	244.63	247.01	14.63		11.53	250.28
1st Stage, " "	101.9	51.53	14.13		9.65	19.63
4th " " "	33.88	18.3	4.02		3.25	6.31
18th " " "	22.75	9.42	2.5		1.94	3.21
Gland Steam, pounds gauge	2.97	1.06	1.93		1.02	2.30
Vacuum, Main Cond., inches of mercury	28.4	29.3	28.25		27.83	28.16
Barometer, inches of mercury	30.54	30.24	30.03		29.83	29.94
Auxiliary Steam, pounds gauge	254.25	240.75	244.38		246.0	243.5
Exhaust, " "	14.68	14.75	8.44		9.75	11.81
Main Feed, at pumps, " "	301.25	313.13	313.13		350.0	328.13
Forced Lubrication, " "	59.06	51.07	45.07		—	48.0
Fuel Oil, pounds gauge	248.0	136.88	153.68		130.0	165.0
Air Pressure in Fire Rooms, inches of water	5.46	4.085	4.38		3.4	4.75
Revolutions or Double Strokes per Min., (average):						
Starb'd. Shaft	569.1	411.75	252.86		194.29	253.53
Port "	571.0	412.35	249.41		196.07	251.02
Mean	570.1	412.05	251.14		195.18	252.6
Main Air Pumps	25.72	26.88	18.5		24.0	15.75
Circulating Pumps	254.15	216.88	196.25		192.02	200.0
Feed Pumps	34.43	37.0	12.75		9.75	15.63
Forced Draft Blowers	1475.8	1180.4	1281.3		1132.5	1500.0
Fuel Oil Service Pumps	32.43	15.94	9.38		6.13	11.5
Shaft Horse Power, (average):						
Starb'd. Shaft	8324	3629	821 ●		337 ●	812
Port "	7984	3537	707 ●		333 ●	729
Total	16308	7166	1528 ●		670 ●	1541
Temperatures, Degrees F. (average):						
Main Air Pump Discharge	78.04	71.07	54.19		47.5	53.57
Injection	47.56	46.75	38.75		36.5	38.5
Outboard Delivery	83.44	71.32	63.0		65.92	60.63
Outside Air	58.75	53.25	44.88		46.0	54.25
Engine Room	88.31	82.25	83.5		79.88	85.75
Auxiliary Room	84.06	79.5	83.65		83.25	88.38
Fire Rooms	82.47	83.26	67.63		71.88	69.25
Feed Water	157.59	185.7	173.5		191.88	168.5
Fuel Oil to Heaters	52.98	53.44	64.0		65.25	61.75
Burners	197.8	175.04	153.88		153.38	177.5
Smoke Pipes	569.2	506.88	490.0		413.75	675.63
Water Consumption:						
Pounds per Hour, measured	301,231	152,860.5	40,745		25,881	55,676
Reserve Feed	2,766.5	1,064.0	1,176		1,999.25	549
Total evaporated	303,997.5	153,924.5	41,921		27,880.25	56,225
per S.H.P.	18.471	21.331	26.666		38.628	36.13
Evap'd. per sq. ft. of H.S.	11.491	7.758	6.338		4.215	8.501
lb. of oil	12.31	13.94	13.13		13.04	12.76
Knot run	10,452.071	6,380.764	2,691.903		2,293.3[illegible]	3,590.8[illegible]
Fuel Oil Consumption:						
B.T.U. per pound	19,436	19,461	19,461		19,461	19,461
Sp. G. at 60° F.	0.8755	0.874	0.874		0.874	0.874
Pounds per Hour	24,465	10,968	3,103		1985	4363
Knot run	841.56	455.24	189.26		163.28	278.66
At	29 Kts. 832	24 Kts. 450.5	15½ Kts. 190		12 Kts. 162	15½ Kts. 276
Guaranteed at	700	460.0	210		175	None
(A) above or (B) below guarantee	(A) 132	(B) 9.5	(B) 12		(B) 13	—
Hour per S.H.P.	1.5	1.531	2.031		2.963	2.831
sq. ft. of H.S.	0.925	0.553	0.469	●	0.3	0.66

NOTE:—Figures in brackets ( ) indicate number of auxiliaries in operation.
● I.H.P., Cruising Engines.—Trial (d), 363 Star., 404 Port, 767 Total; Trial (f) 228 Star., 191 Port, 418 Total.

curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*).

Trial (*a*).—The standardization trial was successfully run on January 5, 1915, on the measured-mile course off Delaware Breakwater. In all 27 runs were made at various speeds. Run No. 19 being unsatisfactory was thrown out. From the data obtained it was found to require 567.6 r.p.m. of the propellers to attain the designed speed of 29 knots; 409.9 r.p.m. for 24 knots, 250 r.p.m. for 15½ knots, and 192.6 r.p.m. for 12 knots.

Table I contains the data from which the curves, Plate I, were plotted.

Trials (*b*), (*c*), (*d*), (*e*), (*f*) and (*g*) were conducted on the dates noted in Table II, which also gives the data obtained.

DESCRIPTION AND TRIALS OF STEAMSHIPS *GREAT NORTHERN* AND *NORTHERN PACIFIC*.

BY W. B. ROBINS, ASSOCIATE MEMBER.

For the purpose of maintaining an express steamer service between San Francisco, Cal., and Astoria, Oregon, the Great Northern Pacific Steamship Company has had built by the Wm. Cramp & Sons Ship and Engine Building Company, Philadelphia, Pa., two ocean-going vessels, the *Great Northern* and the *Northern Pacific*.

These ships have a service speed of 23 knots, and at this speed will make the trip in from 25 to 26 hours, about the same time as it is required for the railway trip between San Francisco and Portland.

To both the naval architect and the traveling public these vessels represent a notable achievement in design and construction, being seaworthy, fast, and provided with every comfort and luxury for passengers. The hull is built on extremely fine lines, while at the same time ample space is provided for 856 passengers, 198 crew, about 2,185 tons of freight, and the 25,000 horsepower propelling plant, with its fuel and auxiliaries.

The passenger accommodations are divided into first, second and third classes, and the arrangement of staterooms, saloons, entrances, deck space, etc., provides for the comfort and convenience of all classes of passengers in all kinds of weather.

In order to get the necessary power for the high speed at which these vessels run, without encroaching upon the space required for passenger accommodation, the type of machinery adopted is a three-shaft Parsons turbine arrangement, together with specially designed oil-burning water-tube boilers.

The machinery arrangement is remarkable for its compactness without being anywhere so crowded as to interfere with efficient operation and attendance. Among the special features of the machinery plant are a system of Pioneer smoke indicators, and the Rites system of lubrication, using a large gravity tank for the oil supply to bearings.

GENERAL ARRANGEMENT.

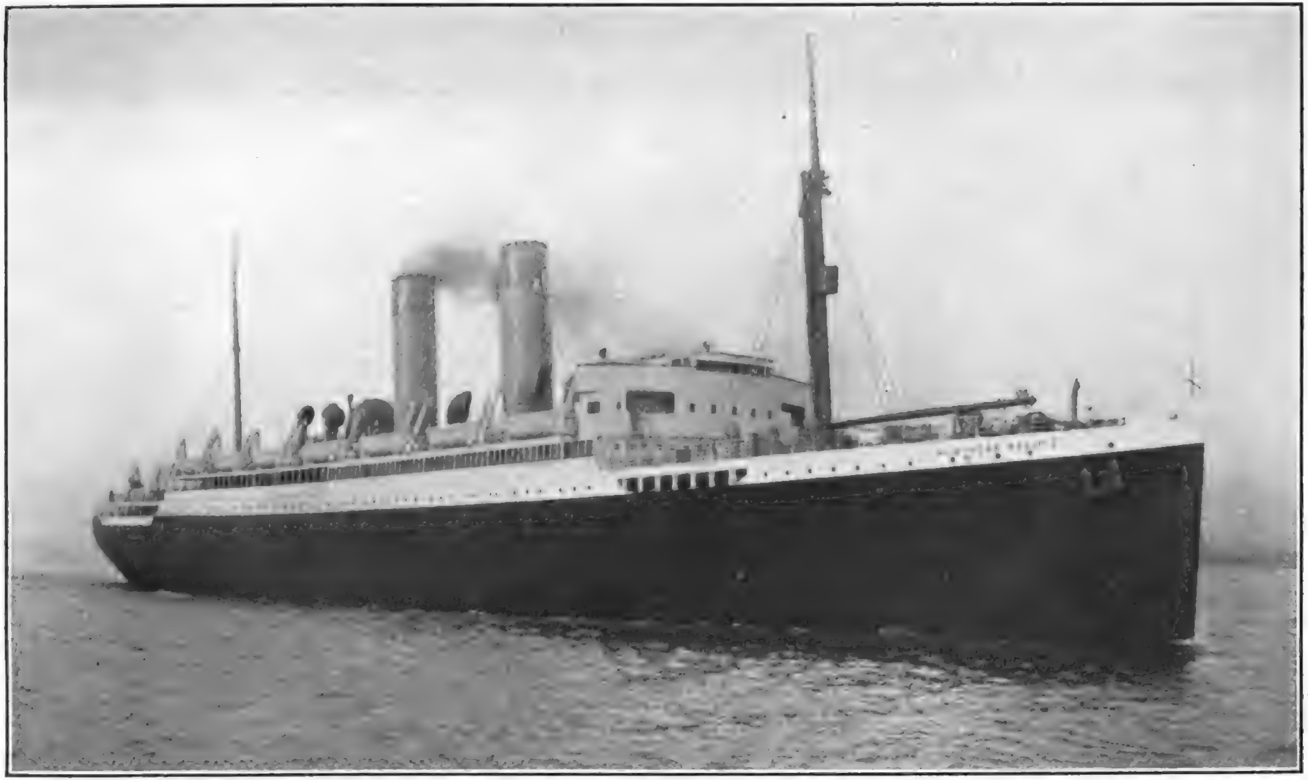
The vessels have straight stems, semi-elliptical sterns, two smoke stacks and two masts. There are six decks and a cellular double bottom, the latter being designed for carrying either fuel oil or water ballast. Beginning at the top, there are the A deck (promenade), B deck (bridge), C deck (shelter), D deck (upper) and E deck (second). Above the A deck there is a boat deck occupying about half the length of the ship. The boat deck is used for stowage of life boats and has a deck forward for officers' quarters, with pilot house and chart room above. The A deck consists principally of promenade space and has a long deck house containing the first-class observation room, smoking room and lobby, and a few staterooms. On this deck, a short distance forward of the deck house, there is a weather bulkhead completely across the ship and the sides are enclosed with sliding glass windows of the Laycock patent type for about 125 feet abaft this bulkhead, and a partly enclosed veranda is fitted up at the after end of the deck house. On the forward end of the A deck are the anchor windlass, 2 capstans and the winches for the forward cargo booms. The main part of the B deck is occupied by first-class staterooms. On the forward part of this deck are the third-class lavatories, dining room, and deck space; and at the after end are the second-class smoking room, lounge and deck space, the hand-steering gear and two capstans. The C deck is almost entirely occupied by staterooms; the first-class amidships, second-class aft and third-class forward. The quarters for the engineer's force are on this deck between the first and second-class passenger quarters. This deck also contains the seamen's quarters

forward, and quarters for the stewards, etc., aft. The steering engine is located on the after end of this deck. The D deck amidships is used for the first and second-class dining saloons, pantries and galleys. At the extreme forward end of this deck are third-class quarters and at the after end are stewards' quarters. There is cargo space on this deck both forward and abaft the main saloons, these spaces being fitted with large doors in the ship's sides. The E deck forward and abaft the machinery space is utilized principally as cargo space. On this deck, abaft the engine room, there is a cold-storage cargo compartment of about 19,460 cubic feet capacity. The hold consists principally of the machinery compartments and cargo space with fuel-oil tanks alongside the boiler rooms. The double bottom is divided in compartments for holding fuel oil and reserve feed water.

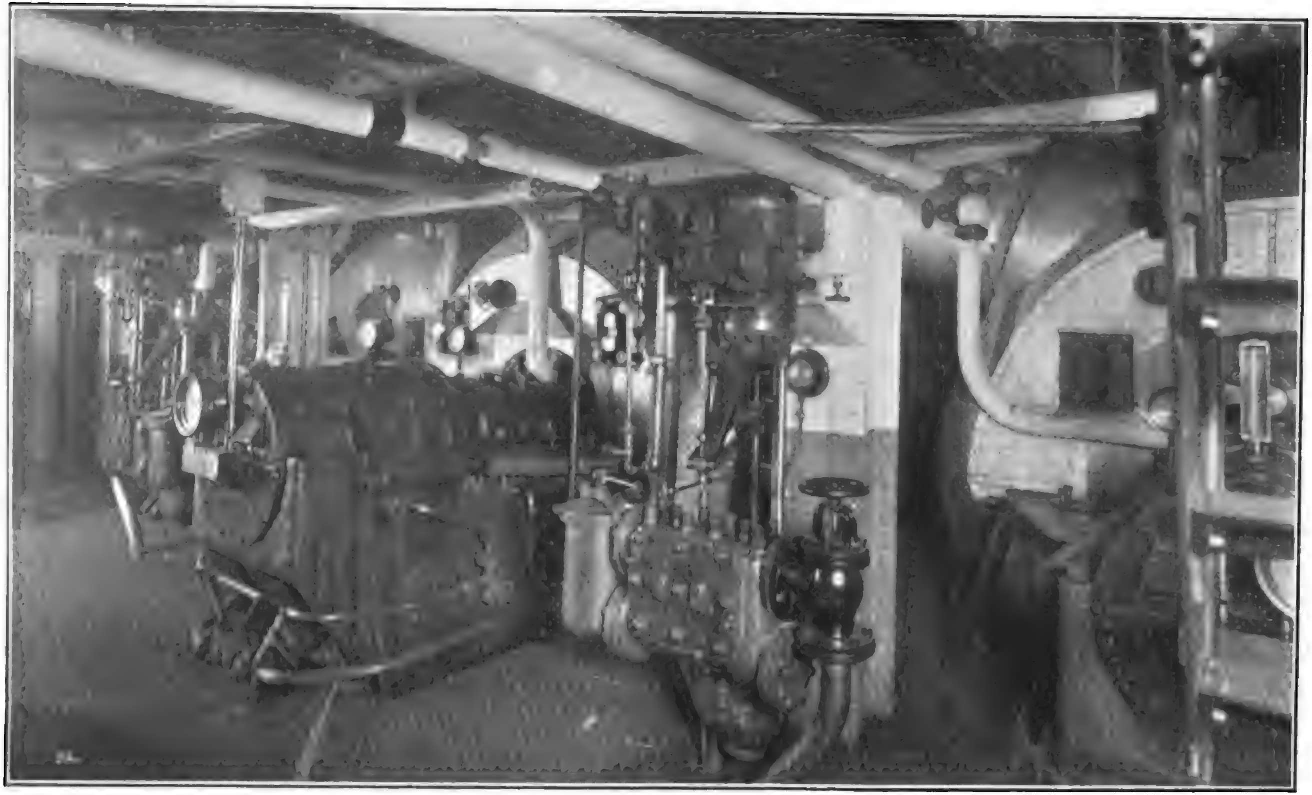
Hull Data.

Length overall, feet	524
between perpendiculars, feet	500
Beam, feet	63
Depth, molded to A deck, feet and inches	50–8
Draught, full load, feet	21
Displacement, tons	9,700
Deadweight carrying capacity, tons	2,185
Gross tonnage, tons	8,255
Capacity of fuel-oil compartments, tons	1,200
reserve-feed tanks, tons	112
drinking-water tanks, tons	102
Passengers, first-class	550
second-class	108
third-class	198
Total	856
Crew	198
Total persons on board	1,054

The stem is of rolled steel fitted to a cast-steel shoe piece, and the stern frame is cast steel. The rudder is solid cast steel with a forged-steel stock. The flat keel is double for three-fifths of its length. Besides the center keel there are three intercostal longitudinals on each side. Above the inner



S. S. "NORTHERN PACIFIC."



Engine Room.—S. S. "Great Northern" and "Northern Pacific."

bottom there is a longitudinal bulkhead on each side extending from the forward boiler room bulkhead to the forward engine room bulkhead. The frames are spaced 30 inches amidships and 28 inches at the forward end. The A deck is completely plated for about one-half the length amidships, and this part of it is a strength deck. The B, C and D decks are completely plated.

Machinery Data.

Shaft horsepower, designed	22,000
Revolutions per minute, designed	350
Speed of ship, designed, knots	23
H.P. turbine, number of	1
length, overall, feet and inches	21–7½
diameter of rotor drum, feet and inches	5–8
number of stages	4
L.P. and astern turbines:	
Number of	2
Length overall, feet and inches	32–2
Diameter of rotor drum, L.P., feet and inches	7–8
Diameter of rotor drum, astern, feet and inches	6–7
Number of stages, L.P.	6
Number of stages, astern	4
Diameter of line shafting, inches	12¼
Propellers, solid manganese bronze, number of	3
number of blades, each	3
diameter, feet and inches	9–2
pitch, feet and inches	8–4
Main condensers, number of	2
cooling surface, each, square feet	13,046
total, square feet	26,092
number of tubes, each	6,018
diameter of tubes, inch	0.¾
thickness of tubes, B.W.G.	16
Boilers, number of	12
width overall, feet and inches	15–7⅜
length overall, feet and inches	11–8 7/16
height over steam drum, feet and inches	14–2½
diameter of steam drum, inches	54
number of tubes, each	969
diameter of tubes, inches	2
thickness of tubes, B.W.G.	9
heating surface, each, square feet	5,000
total, square feet	60,000
working pressure, pounds	220

The turbines are the Parsons type, arranged on three shafts, one high-pressure on the center shaft and one combined low-pressure and astern on each wing shaft. The turbine casings are of close-grained cast iron, the rotor drums are weldless-steel forgings. The drum wheels are steel castings and the rotor shafts are steel forgings. Lubrication of the turbine and main shaft bearings is by a gravity system. A tank of about 2,000 gallons, capacity is placed at height, which gives a pressure at the bearings of about 10 pounds per square inch. After the oil has passed through the bearings it flows to drain tanks from which it is returned to the gravity tank by two steam pumps, two coolers being fitted to lower the temperature of the oil.

The boilers are of the Mosher type, built by the Babcock & Wilcox Company, and are arranged in two compartments, six in each. The boilers are all set facing the center of the ship, leaving a firing space along the center line 11 feet 2½ inches wide from front to front of boilers. The total space occupied by the boilers is 130 feet x 41 feet 6 inches. At the forward end of the forward fireroom there is a donkey boiler 5 feet 9 inches, diameter, and 10 feet 7¾ inches long. The boilers use oil fuel, and operate under forced draft. The Koerting mechanical oil-burning system, made by the Schutte and Koerting Company, Philadelphia, Pa., is used. The principal features of this system are forcing the oil under high pressure through heaters and atomizing it mechanically in centrifugal spray-type burners without the use of steam or air as an atomizing agent, the air for combustion being furnished from the forced-draft pressure in the fireroom through an adjustable air register at each burner.

There are two smoke stacks, about 112 feet high above the base line. The Pioneer smoke indicator system is installed, which enables the density of smoke passing up the stacks to be determined in the fireroom. This system consists of an electric light in each smoke stack shining through a lens onto a selenium cell on the opposite side of the stack. The selenium cell is in an electric circuit, and a variation in the intensity of

the light changes the conductivity of the selenium, causing a variation in the current, this variation showing on an indicator in the fireroom.

The forced draft is operated on the closed-fireroom system, three 34-inch high-speed turbine-driven cone fans being fitted for each fireroom. These blowers are made by the B. F. Sturtevant Company, Hyde Park, Mass., and have a capacity of 30,000 cubic feet of air per minute against a pressure of 2 inches of water.

The following pumps were furnished by the Blake & Knowles Steam Pump Works:

2 Main air, vertical, twin-beam, single-acting, 14 x 32 x 21 inches.
1 Auxiliary air, vertical, featherweight simplex, 7½ x 14 x 10 inches.
4 Main feed, vertical simplex, 15½ x 9¾ x 24 inches.
2 Auxiliary feed, vertical duplex, 12 x 8½ x 12 inches.
2 Bilge and ballast, vertical duplex, 7½ x 9 x 10 inches.
2 Lubricating oil, vertical simplex, 6 x 8 x 12 inches.
4 Fuel-oil service, horizontal duplex, 7½ x 5 x 10 inches.
2 Fuel-oil transfer, vertical simplex, 6 x 8 x 12 inches.
2 Brine, vertical duplex, 5¼ x 5 x 5 inches.
1 Donkey boiler feed, horizontal simplex, 4½ x 2¾ x 6 inches.
1 Evaporator feed, horizontal simplex, 4½ x 2¾ x 6 inches.
2 Fresh-water, horizontal simplex, 5½ x 5½ x 7 inches.
1 Ice-water circulating, horizontal simplex, 3½ x 3½ x 4 inches.

Besides the above there are two 4-inch Worthington centrifugal sanitary pumps with a capacity of 500 gallons of salt water per minute against a head of 100 feet. These pumps are driven by a 20-horsepower Terry steam turbine, which runs at 2,750 revolutions per minute with initial steam pressure of 175 pounds and back pressure of 10 pounds. There are two 48-inch Reilly multicoil feed-water heaters, each having a capacity of 150,000 pounds of water per hour with a discharge temperature of 208 degrees F., with steam at 5 pounds pressure. There is also a Reilly multicoil Navy type evaporator, having a capacity of 25 tons of fresh water per 24 hours.

ELECTRIC PLANT.

Electric current is furnished by four 35 kw., 110-volt generators made by the Diehl Manufacturing Company. These generators are direct-connected to type "G M" Terry steam turbines, designed to operate with 200 pounds steam pressure and vacuum of 28 inches, but having sufficient power to deliver the rated output of the generators with 175 pounds steam pressure and a 26-inch vacuum; the normal running speed being 3,200 revolutions per minute. These generators supply current for lighting, etc., for operating five freight elevators and for operating a Willets-Bruce automatic whistle.

REFRIGERATING PLANT.

The refrigerating plant is the CO_2 type made by J. & E. Hall Company, Ltd. There are two machines having sufficient combined capacity to refrigerate the cargo compartment of 19,462 cubic feet and the provision compartments of 2,778 cubic feet, maintaining a temperature of 20 degrees F., when the outside temperature is 85 degrees F.

DECK EQUIPMENT.

The steam-steering gear is of the Williamson differential lever type with 14-inch x 14-inch cylinders, operated by telemotor from the bridge or direct at the engine. This gear, as well as the windless, capstans and deck winches, was made by the American Engineering Company, Philadelphia. There are three bower anchors of 9,000 pounds each; one stream anchor of 2,500 pounds, and one kedge of 1,200 pounds. There are two 2½-inch chain cables, each 150 fathoms long, and one 7-inch hawser, 120 fathoms long, for the stream anchor.

The following small boats are furnished: Thirteen metallic life boats, 28 feet x 9 feet 4 inches x 4 feet 1½ inches, 64 persons each; four collapsible life boats, 29 feet x 9 feet 10 inches

x 3 feet, 54 persons each; one metallic working boat, 22 feet x 6 feet x 2 feet 3 inches. The boats are handled by Horton sheath davits.

FIRE-ALARM SYSTEM.

The Aero Automatic Fire Alarm System is installed throughout the principal compartments of the ship. Each unit of this system consists of a small copper tube leading from a switchboard through its compartment and back to the switchboard. Any rapid rise of temperature in the compartment causes expansion of the air in the tube which operates a sensitive diaphragm at the switchboard, closing an electric circuit which rings a fire bell in the engine room and a buzzer at the switchboard. There are three divisions of the system, the three switchboards being located forward, amidships and aft on the C deck. There are two main annunciator boards, one in the engine room and one in the pilot house. The annunciators are in the form of diagrams of the ship, and the alarm shows in what part of the ship and on which deck the fire is located. The division of the system into three systems reduces the danger of panic, as an alarm is given only on the switchboard for its part of the ship, in the engine room and in the pilot house, thus giving an opportunity to put out the fire without alarming the passengers in the other parts of the ship.

TRIALS.

The contract required the following:

(*a*) A progressive standardization trial over the measured mile to standardize the propellers.

(*b*) A 12-hour trial at 20 knots; the revolutions to be taken from the revolution curve obtained by (*a*); and further, on this trial ten of the twelve boilers shall be in use, and four of the six forced-draft blowers.

The S. S. *Great Northern* was the first vessel ready for trial. It was decided to standardize over the measured mile at Rockland, Maine.

The standardization trial was conducted in December, 1914,

and the curve in Plate I obtained. Immediately after the three high-speed runs, the vessel began her 12-hour, 23-knot run. The speed was easily maintained and the performance of the machinery was excellent. Data is given in Table I.

En route to Rockland, the *Great Northern* encountered a heavy N. E. gale, and the owners were given an opportunity to ascertain her sea-going qualities. Accordingly the vessel was run at various speeds, up to full speed, with the sea ahead, on the bow and beam. The maximum pitch was 4 degrees to 5 degrees, and the maximum average roll 18 degrees to 20 degrees. The vessel was remarkably steady and very dry forward.

The S. S. *Northern Pacific* underwent her trials in February, 1915, and her performance was most excellent. Her average speed was slightly higher. The data is given in Table II.

One of the most excellent qualities of these vessels is the entire absence of vibration at high speed in the passenger quarters. There is only a slight vibration over the propellers.

In the engine design of the vessel there are several noteworthy ideas.

The arrangement in the engine room and firerooms is par-excellent. In the engine room it is possible to stand on the throttle platform and observe the operation of every piece of auxiliary machinery. This feature is extremely valuable to merchant service, where the maximum results must be obtained from the minimum cost.

The arrangement of the oil coolers under the sanitary pumps, the foundations for the pumps being cast with the cooler castings, and the water passing through them to the sanitary system, is one deserving of attention as to what can be saved in weight and space with a gain in efficiency.

The turbine forced-lubrication system is about the same as the standard Parsons system, except the pressure is maintained by the height of the supply tank in the engine-room hatch. There is an oil-level sight glass on the working platform and also a float in the tank, so that as soon as the oil drops to a

12 HOUR – 23 KNOT RUN. DEC. 12, 1914. 8:10 A.M. TO 8:10 P.M. F

ENGINE ROOM DATA.

	TIME	A.M. 8.10 TO 9.30	A.M. 10.30	A.M. 11.30	P.M. 12.30	P.M. 1.30	P.M. 2.30	P.M. 3.30	P.M. 4.30	P.M. 5.30	P.M. 6.30
	R.P.M. FOR PERIOD P.	363.7	3383	3440	346.7	3474	3471	3492	350.5	3488	3435
	DO. C.	336.4	320.5	325.5	3279	3287	3286	3301	331.3	329.3	3278
	DO. S.	358.0	3339	338.7	3414	3426	3424	344.5	345.7	3439	3379
	AVERAGE	352.7	3309	336	3387	3396	3394	3413	342.5	340.7	3364
	RPM FOR TORSION METER CARDS P.										
	DO. C.										
	DO. S.										
	AVERAGE										
	S.H.P. P.										
	DO. C.										
	DO. S.										
	TOTAL										
PRESSURES	MAIN STEAM P.		205	203	208	203	201	210	208	196	210
PRESSURES	DO. S.		202	202	205	202	201	209	206	196	208
PRESSURES	MAIN STEAM ENGINE ROOM		198	198	202	196	198	203	200	190	202
PRESSURES	MAIN H.P. AHEAD		160	162	165	162	160	169	167	158	172
PRESSURES	L.P. AHEAD P.		19	25	26	20	20	22	21	19	22
PRESSURES	DO. S.		24	24	25	25	24	26	25	23	26
PRESSURES	VACUUM MAIN CONDNSR P.		29.3	29.3	29.3	28.9	28.9	28.8	28.7	28.8	28.8
PRESSURES	DO. S.		29.6	29.6	29.5	29.5	29.5	29.3	29.3	29.	29.4
PRESSURES	AUXILIARY STEAM		205	208	210	205	202	214	210	199	212
PRESSURES	" EXHAUST		4	4	4	3	4	4	3.5	3.5	4
PRESSURES	STEAM TO GLANDS		1.5	1.5	1.75	1.5	1.5	2	1.9	1.	2.25
PRESSURES	EXHAUST TO FEED HEATER P.		6	1	1	1	1.25	2	2	2	1
PRESSURES	DO. S.		4	—	—	3	—	.5	.5	—	—
PRESSURES	FEED PUMP DISCHARGE P.		270	265	260	265	270	265	270	270	270
PRESSURES	DO. S.		260	265	265	270	265	270	270	270	270
PRESSURES	FEED LINE		268	253	262	255	253	269	261	254	255
TEMPERATURES F°	TEMPERATURE OF INJECTION		45.	46	46	46	46	46	47	47.5	48
TEMPERATURES F°	OVERBOARD DISCHARGE P.		73.5	74	74	74.5	74.5	74.5	75	76	77
TEMPERATURES F°	DO. S.		74	74	74	74	74	74	76	76	76
TEMPERATURES F°	AIR PUMP DISCHARGE P.		56.5	55	55	56	56	56	56	55	59
TEMPERATURES F°	DO. S.		58	57	54	57	57	57	58	55	60
TEMPERATURES F°	FEED TO HEATER P.		83	74	76	75	76	77	80	73	86
TEMPERATURES F°	DO. S.		84	86	77	82	84	80	83	74	88
TEMPERATURES F°	FEED FROM HEATER P.		218	184	180	198	188	197	206	180	196
TEMPERATURES F°	DO. S.		176	208	188	184	198	194	200	170	187
TEMPERATURES F°	WORKING PLATFORM		91.	91.5	94	94	94	93	88	88	87
DS & R.P.M.	R.P.M. CENTRIFUGAL PUMP P.		123	125	127	126	125	126	126	128	126
DS & R.P.M.	DO. S.		122	120	128	127	128	126	126	127	126
DS & R.P.M.	AIR PUMP P.		19	19.5	19.5	13	12	13.5	12.5	11	12.5
DS & R.P.M.	DO. S.		13	12	12	12	12	10	10	10.	10
DS & R.P.M.	FEED PUMP O.P.		9.8	14.3	15.2	8.7	3.9	6.4	6.4	6.6	6.2
DS & R.P.M.	DO. I.P.		4.9	9.7	10.5	14.1	16.8	17.2	18.1	18.4	17.1
DS & R.P.M.	DO. I.S.		12	9	12	12	15.3	17.1	17.8	17.9	18.0
DS & R.P.M.	DO. O.S.		19.3	14.6	14.2	14.5	13.0	8.5	8.9	8.9	8.9
DS & R.P.M.	BILGE & BALLAST PUMP P.		30	44	41	—	51	39	68	73	62
DS & R.P.M.	DO. S.		—	—	—	—	—	—	—	—	—
	LUBRICATING OIL PUMP P.		6	7	7	6.5	6.5	6	6	6	10
	DO. S.		12	12	12	12	12	13	13	11.5	12
LUBRICATING OIL PRESSURES	DISCH. LUB. OIL PUMP P.		27	35	34	32	33	32	32	31	36
LUBRICATING OIL PRESSURES	DO. S.		33	34	35	33	33	35	33.5	32.5	37
LUBRICATING OIL PRESSURES	OIL THRUST BEARING P.		10	9	10	10	10	9.5	9.5	9	10
LUBRICATING OIL PRESSURES	DO. C.		7.5	8	7.5	7	7	8	7.5	7	8
LUBRICATING OIL PRESSURES	DO. S.		9	8.5	8.5	9	8.5	8.5	8	8.5	9
LUBRICATING OIL PRESSURES	OIL FORD. MAIN BEARING P.		9	10	9.5	9	9	10	9.5	9	10
LUBRICATING OIL PRESSURES	DO. C.		9.5	9.5	10	9.5	9	9	9	9	10
LUBRICATING OIL PRESSURES	DO. S.		10	9	10	9	9	9	9	9	10
LUBRICATING OIL PRESSURES	OIL AFTER " " P.		9.5	9	9	9	9	9	8.5	8.5	9
LUBRICATING OIL PRESSURES	" " " " C.		10.5	10.5	10	11	10	10	10	9.5	10
LUBRICATING OIL PRESSURES	" " " " S.		7	7	7	7	7	7	7	7	7.5
LUBRICATING OIL TEMPERATURES F°	OIL DISCH. THRUST BEARING P.		98	100	100	101	100	100	100	100	100
LUBRICATING OIL TEMPERATURES F°	DO. C.		92	94	94	94	94	94	94	94	95
LUBRICATING OIL TEMPERATURES F°	DO. S.		101	102	103	102	103	103	103	103	103
LUBRICATING OIL TEMPERATURES F°	" " FORD. MAIN " P.		104	105	106	106	106	106	105	105	107
LUBRICATING OIL TEMPERATURES F°	DO. C.		97	98	99	98	98	98	98	98	99
LUBRICATING OIL TEMPERATURES F°	DO. S.		105	105	106	106	106	105	105	105	104
LUBRICATING OIL TEMPERATURES F°	" " AFTER MAIN " P.		103	106	107	106	106	105	105	105	105
LUBRICATING OIL TEMPERATURES F°	DO. C.		97	99	100	97	96	95	95	96	100
LUBRICATING OIL TEMPERATURES F°	DO. S.		106	106	107	107	107	107	107	108	107
LUBRICATING OIL TEMPERATURES F°	LUB. OIL DISCH. FROM COOLER		82	83	83	82	82	82	82	83	84
	PRESSURE OF OIL SYSTEM		9	9	9.2	9	9	8.5	8.5	8.25	8.5

8.10 A.M. BEGINNING 1ST STANDARDIZATION RUN AT 23 KNOTS AND ALSO BEGINNING OF 12 HOUR 23 KNOT RUN.
9.20 A.M. FINISH 3RD & LAST STANDARDIZATION RUN AT MAXIMUM SPEED.
RECORDS FOR OIL, ETC., BEGIN AT 9.30 A.M.

T TRIAL.

		A.M. 1.30	P.M 1.00	P.M. 1.30	P.M. 2.30	AVERAGE
	TIME					
	RPM. FOR	212	216	213	215	214
	DO.	215	212	213	217	214
	DO.	213	210	213	217	213
	AVERAGE	217	215	217	218	217
	RPM. FOR TO	216	212	218	217	216
	DO.	216	214	217	217	216
	DO.	208	212	208	210	210
	AVERAGE	207	212	208	213	210
	S. H. P.	213	215	212	215	213
	DO.	208	210	208	210	209
	DO.	208	208	205	208	246
	TOTAL	207	207	205	208	207
PRESSURES LBS PER SQ INCH & INS MERCURY VAC	MAIN ST	1300	1320	1300	1280	1292
	DO.	1340	1350	1340	1350	1344
	MAIN STEA	1300	1280	1300	1320	1260
	MAIN H.P.	1480	1480	1480	1420	1264
	L.P. AHE					
	DO.	1400	1440	1460	1480	1404
	VACUUM					
	DO.					
	AUXILIAR					
	DO.	6	9	9	8	7.5
	STEAM T					
	EXHAUST	13	11	11	11	12
	DO.	215	212	215	215	214
	FEED PUM	215	213	218	218	216
	DO.	2.2	2.3	2.5	2.5	2.3
	FEED LI	2.2	2.2	2.3	2.3	2.2
TEMPERATURES F°	TEMPERA	230	240	240	250	258
	OVERBOA	230	240	230	250	232
	DO.					
	AIR PUM					
	DO.	190	180	190	190	188
	FEED TO	180	180	190	190	186
	DO.	190	180	180	190	186
	FEED FRO	180	180	190	190	186
	DO					
	WORKING	52	53	53	53	52.6
D.S. & R.P.M.	R.P.M. CENT	610	520	500	480	512
	DO.	510	510	510	570	510
	AIR PUM	5	5	6	6	5.4
	DO.	5	3	5	5	4.6
	FEED PU	6	6	6	6	5.8
	DO.	6	6	5	5	5.4
	DO.	6	6	6	6	6.
	DO.	6	6	6	6	6.
	BILGE & BA	4	4	5	4	4.6
	DO	4	4	5	4	4.4
	LUBRICATI	5	5	5	5	5
	DO.	5	5	4 5	5	4.9
LUBRICATING OIL PRESSURES LBS. PER SQ. IN. G.	DISCH. LU	5	5	5	5	5
	DO.	5	5	5	5	5
	PRESSURE	1.3	1.05	1.6	1.9	1.5
	OIL THRUS					
	DO.					
	DO.					
	OIL FORWA					
	DO.					
	DO.					
	OIL AFTER					
	DO.					
	DO.					
LUBRICATING OIL TEMPERATURES	OIL DISCH.					
	DO.					
	DO.					
	DO.					
	DO.					
	DO.					
	DO.					
	DO.					
	DO.					
	LUB. OIL DIS					

OFF DELAWARE CAPES.
METER 29.93
NORTHERN CURVE)
NT 9500 TONS.
AWARE BREAKWATER
TER CLEAR & BRIGHT
3475
COURSE AND WAS ONLY
ATIVE RESULTS BETWEEN
GREAT NORTHERN WAS RUN
0" DRAFT EVEN KEEL 20 FT 5½"
0" DISPLACEMENT 9612 TONS

STANDARDIZATION

WEATHER - CLEAR AND COL		
		RUN NO.
		TIDE
		TIME BEGINNING RUN
		SPEED IN KNOTS
		R.P.M. FOR RUN
		DO.
		DO.
		DO. MEAN
		SLIP %
		S.H.P. FOR RUN
		DO.
		DO.
		DO. TOTAL
	PRESSURES - LBS PER SQ. INCH & INS. MERCURY VAC.	MAIN STEAM LINE
		DO.
		MAIN STEAM ENGINE ROO
		MAIN H.P. AHEAD
		MAIN L.P. AHEAD
		DO.
		VACUUM MAIN COND'N'S'R
		DO.
		AUXILIARY STEAM
		" EXHAUST
		STEAM TO GLANDS
		EXH. TO FEED HEATER
		DO.
		FEED PUMP DISCHARGE
		DO.
		FEED LINE
	TEMPERATURES - F	INJECTION
		OVERBOARD DISCHARGE
		DO.
		AIR PUMP DISCHARGE
		DO.
		FEED TO HEATER
		DO.
		FEED FROM HEATER
		DO.
		WORKING PLATFORM
	D.S. & R.P.M.	R.P.M. CENTRIFUGAL PUMP
		DO.
		AIR PUMP
		DO.
		FEED PUMP
		DO.
		DO.
		DO.
		BILGE & BALLAST PUMP
		DO.
		LUBRICATING OIL PUMP
		DO.
LUBRICATING OIL	PRESSURES LBS PER SQ. IN. GAGE	DISCH. LUB. OIL PUMP
		DO.
		LUBRICATING OIL SYSTEM
		OIL THRUST BEARING
		DO.
		DO.
		OIL FOR'D MAIN BEARING
		DO.
		DO.
		OIL AFTER " "
		OIL AFTER " "
		DO.
	TEMPERATURES F	OIL DISCH. THRUST BEARING
		DO.
		DO.
		" " FOR'D MAIN "
		DO.
		DO.
		" " AFTER MAIN "
		DO.
		DO.
		LUB. OIL DISCH FROM COOLER
		NO. OF BOILERS IN USE
		BOILER PRESS. AVERAGE
		NO. & SIZE OF BURNERS TOTAL
		R.P.M. BLOWERS & BLOWERS IN US
		AIR PRESS. BOILER ROOM & ATMO
		DO. UPTAK
		PRESSURE OIL TO BURNERS
		TEMPERATURE DO.
		NO. & D.S. OF FUEL OIL PUMPS

DESIGN
LENGTH, P.P.S. 500 FT - 0"
BEAM MOLDED 63 FT - 0"
DRAFT EVEN KEEL 20 FT 5¼"
DISPLACEMENT 9612 TONS

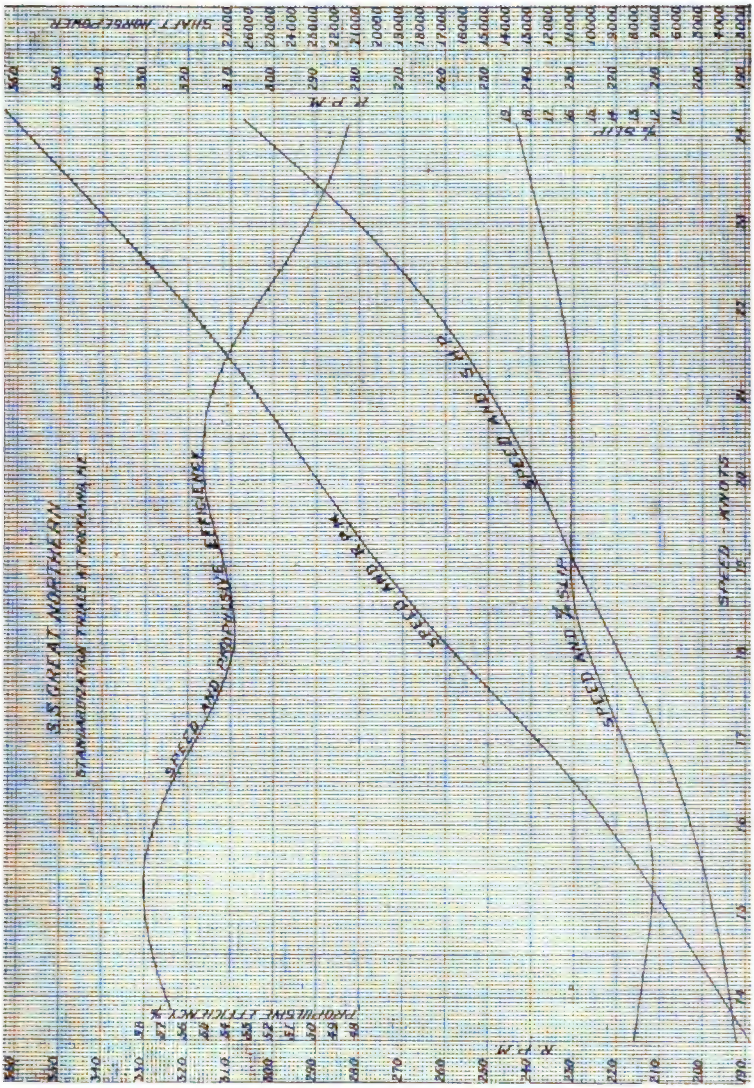


PLATE I.

low level a large 12-inch gong sounds. This prevents, by any chance, the oil level getting low. The oil is returned to the tank by a pump automatically controlled by a pressure governor of the "Ideal" make.

In the stern-tube lubrication there is another feature new to this country, viz: the Vickers oil-packed stern tube. This system has been in use for some time on the Channel steamers in England, it being made by the Vickers firm of England. It is claimed to reduce friction and to prevent the wearing down of the lignum vitae packing in the stern tubes.

The system consists practically of filling the stern tube with a special oil mixture, the composition of which is held by the Vickers Company, and maintaining a head on this by means of a tank located at a height sufficient to make the head of oil on the stern tube just slightly greater than that of the water on the outside.

The operation of the system on the trials was excellent. In connection with this system the Cramp Company deserve credit for installing this system, as these vessels are by far the largest at present using it.

The fireroom arrangement is excellent. There are several novel and very excellent features in it. First, the boilers are a new type and gave very excellent results. There are very few joints to care for, and at the same time have the excellent feature of a large combustion chamber, and the same gas passage as a standard Navy B. & W. boiler. It is believed the boilers will require very little upkeep, as compared with the standard B. & W. boiler.

Outside of this there are two novel features in the boilers: (1) The feed is introduced into the lower or mud drum, and the burners are under this drum. (2) The air is admitted in two ways: First, from the back of the boiler, down, and under the boiler floor and up to the burners in front, and also through doors at the front. This resulted in the boiler castings being remarkably cool.

The Pioneer Smoke Indicator System is also installed. It operated excellently, and gives the density of the gas and also an approximate CO_2 analysis, thus enabling the fireman to maintain an excellent combustion, and also to control the smoke nuisance. This feature is also new, this being the first

installation of its kind, and is somewhat of an experiment, as the device is not yet on the market.

The vessels are the finest merchant vessels ever built in this country and are in many ways far ahead of anything foreign builders can boast of. They are a great credit to the builders and a valuable addition to the American merchant service.

SUBMARINES AND TORPEDOES.

By Lieutenant (J. G.) C. N. Hinkamp, U. S. Navy, Member.

Interest in the submarine has reached a point hardly dreamed of five years ago. This has been brought about by the prominent part played by the submarine in the present war in Europe and by the recent catastrophe involving one of the submarines of the United States Navy.

Reviewing briefly the history of the development of submarines, the first attempt at submersible devices is found in the early diving bells. Records show these to have been used long before the Christian era. As far as is known, the first successes occurred about 1620. The first submarine was a vessel built of wood and covered with leather. Since that date, 1620, governments and individuals have been developing submarines.

In 1773 Bushnell began the construction of a submarine. It was of copper and made of two shells. The features in his boat were the arrangement of torpedoes, lead keel, ballast tanks and pumps for freeing these tanks.

The French began early to experiment with submarines, but until about 1797 very little success was obtained by them. Robert Fulton, an American, entered the field in 1800, and in 1801 began to get some satisfactory results with an experimental boat operating in the river Seine.

In 1850 William Bauer, a German, drew plans for a boat, and in 1856 built one for Russia. This boat was propelled by treadmills. It was lost, however, and no more of the type were built.

The first period in which submarines were actually employed in naval service was during the American Civil War, and no

really successful operation of this type of vessel is found until that time.

In 1863, a submarine designed by Alstill was built in Mobile, Alabama. This boat was not used, but had several features in its design which are still considered excellent.

The next series of boats were classed as "Davids." They ran partially exposed and could hardly be called submarines. The Davids were more or less successful; some were lost, but their successes in the Civil War paved the way to the weapon of the present day.

In 1863 the French again took up the problem. Their vessels were designed to move along the bottom, but very little actual experimenting was done with them. Their success was indifferent.

In 1881 the Nordenfeldts took up the design of submarines, and in 1885 built a vessel for the Turkish Government. The propulsive power for this boat was steam.

In 1888 the Goubet submarines appeared, with indifferent success.

The next design appeared in 1897 and was the creation of M. Laubeuf. This type was really the first example of the modern submarine, so it may be said that the date of modern submarine construction is 1897, or just prior to the Spanish-American War.

The inventors in the United States were not idle during this time. The two famous American inventors, Holland and Lake, were each perfecting a type of boat and these types have now been adopted as practically the standards of the world. Holland's experiments began in 1875, but it was not until 1893 that his plans were approved by the American Naval authorities. In 1904 Lake's boats were built for Russia.

Up to 1898 the submarine was in the embryonic state. From then up to the present time it has developed rapidly and great strides have been made in its design with advances in the military features as well as sea-keeping qualities.

In the United States the development has been rapid. The present appropriation for the building of submarines calls for

two distinct types, sea-going, of 1,200 tons, and coast or harbor-defense submarines, of about 400 tons. The contracts for these have recently been awarded. These types present added features making for increased comfort for the crew and embody many features to increase the military efficiency.

In the construction of a submarine, consideration has to be given to two kinds of tanks, namely, ballast tanks and trimming tanks. The ballast tanks are divided into main ballast tanks and auxiliary ballast tanks. The trimming tanks are tanks in the bow and stern of the boat used for trimming the vessel in the fore-and-aft line. The ballast tanks destroy the maximum part of the reserve buoyancy when completely filled. The smaller ballast tanks are used finally to trim the vessel to a predetermined amount of buoyancy. In addition to the ballast and trimming tanks the vessel is fitted with fuel tanks to carry the necessary fuel for the main engines.

The torpedo tubes are located in the bow of the submarine. In some classes there are surface tubes and tubes in the stern, but ordinarily they are fitted only in the bow. In any case, it is necessary to aim the boat whenever a torpedo is to be fired.

The engines for propelling the vessel on the surface are located in the stern as are also the main motors.

The storage batteries for submerged navigation are generally located amidships.

In the following an attempt is made to describe the use of this combination of tanks, engines, batteries and torpedoes.

It should be understood that there is nothing mysterious in the operation of a submarine. The orders used in the handling of a boat are few; they are made as comprehensive as possible and are so given as to eliminate any possible confusion.

Preparing to submerge includes all preliminary work up to the closing of the conning-tower hatch. This comprises the stowing of the deck gear, taking down the bridge, unrigging the radio, closing the hatches, unlocking the valve-operating mechanism, securing the engines; in fact, a clearing ship for action. This operation requires from two to twenty minutes, depending on the amount of rigging to be taken down.

Before proceeding with the description of the actual submerging of a boat, a few definitions of terms used will be given:

Positive Buoyancy.—That condition in which a boat any part of the structure of which is above water requires ballast to sink.

Negative Buoyancy.—That condition in which a submerged boat without headway requires the removal of ballast to rise.

Neutral State.—That condition in which a completely submerged vessel is in equilibrium.

Submerged Trim.—The amount of positive or negative buoyancy a vessel possesses with reference to the neutral state.

Awash Condition.—That condition in which the main ballast tanks are empty and the auxiliary ballast and trimming tanks contain the requisite ballast so that by flooding the main ballast tanks the vessel will have the proper trim for submerged operations.

Flooding.—That process whereby water is admitted to a tank directly from the sea.

Filling.—That process whereby water is pumped into a tank.

Blowing.—That process whereby water is forced out of a tank by means of air pressure.

Pumping.—Pumping water overboard from the tanks by means of main or auxiliary pumps.

The actual submerging of the boat can be done in two ways, one called the "static" dive, the other the "running dive."

In the static dive, also known as "balancing," the boat is submerged but does not move except in the vertical plane. This dive may be accomplished in two ways: by trimming the boat and maintaining her trim by adjusting the ballast, or by dropping the anchor, trimming the boat to within a few hundred pounds positive buoyancy, and the heaving in or veering on the anchor cable. The latter way is the simpler method for easy control and can be used where there is no current, or only a small amount of current, if the sea is not too rough.

The principles involved in balancing a submarine are merely those by which is determined the force of gravitation. The

mass of the submarine itself amounts to several hundred tons, but the actual forces used to sink it from a neutral state are very small. The addition of 50 pounds of water will cause the boat slowly to descend. As soon as the vessel has started on its downward path it is necessary to remove part of the ballast in order to keep it from continuing on this path, provided the density of the water is constant. By pumping out, or blowing, a given quantity of ballast, there will be a force exerting an upward tendency. The determination of the amount of ballast necessary, the forces, is the difficult part of balancing. In balancing by the use of the anchor, the different forces do not have to be considered to such a great extent.

Before submerging, the vessel is generally brought to a fore-and-aft trim which will cause the boat to be level when submerged. This is done by flooding or filling the forward or after trimming tanks.

Balancing by means of the anchor is, roughly, accomplished as follows: The anchor, which weighs about 1,000 pounds, sometimes more, is dropped. The boat is then trimmed down until it is in practically a neutral state, or has an amount of buoyancy equivalent to two to three hundred pounds. It can readily be seen that with a reserve buoyancy of three hundred pounds and a weight of a thousand pounds on the bottom there is a preponderance in favor of the anchor of about six hundred to seven hundred pounds. By slowly heaving in on the anchor cable, the positive buoyancy of the boat is easily overcome and she may be drawn down to any depth desired.

The static dive by adjustment of ballast is made as follows: After getting the fore-and-aft trim, as already described, the main ballast tanks are flooded. These tanks are flooded through large kingston valves, the operation requiring from one to two minutes. There is an enormous volume of air in the ballast tanks that must be freed before the tanks can fill with water, and the process of getting rid of this air is called "venting." The main ballast tanks "vent" overboard, as the large volume of air, if admitted into the interior of the boat, would cause the barometric pressure in the boat to rise to such

a point as to be uncomfortable. All the other tanks, being comparatively small, vent inboard.

The boat being trimmed down as far as the main deck, still has too much buoyancy to run submerged. The tank next flooded is the auxiliary ballast tank. This tank holds enough water to destroy the remaining buoyancy and is flooded generally until the top of the conning tower is just under the surface of the water. The final trimming is done by slowly flooding or filling the adjusting tank. When the vessel is trimmed until there is about two to three hundred pounds of positive buoyancy, it can be readily handled submerged. This is considered the best trim for all around work, and completes the static dive. From this condition any operation submerged can be commenced.

The state of the sea affects balancing. In trimming, the boat oscillates until it strikes a condition of comparative equilibrium, when it comes to rest. With a " sea" on, this state of equilibrium is never reached, and the amount of water taken in to approach a neutral state is difficult to estimate owing to the fact that the amount of positive buoyancy depends on the area of the water-line plane which varies with every wave that passes. Suppose a vessel rising to the effect of a passing wave adds a little water in the ballast tank. As soon as the wave passes, the boat has a tendency to sink and, with the added impetus caused by additional ballast, it falls rapidly. If the " downward send" is sufficient, the only thing that will stop the vessel is the bottom. The downward velocity may be even great enough to overcome a fairly large amount of buoyancy.

To be able to determine when to stop is almost the entire secret of the art of balancing. For strategical purposes it is better to have a small amount of headway on the boat, just enough to overcome the effect of the sea which, being rough, would hide the wake of the periscope in the foam of the white caps.

The running dive is made from the awash condition. In the awash condition the trimming tanks and auxiliary ballast tanks are flooded to the amount necessary for the proper trim

when submerged; the main ballast tanks are empty. The running dive is used for all tactical purposes except balancing.

The vessel being underway "awash," the order is given to submerge. All hands get into the boat, the engines are stopped and the electric motors started. As soon as the engines are stopped the conning tower is closed, all ventilators housed and the main ballast tank flooded. Knowledge that the trim will be approximately correct when totally submerged renders careful adjusting of ballast unnecessary. The boat is inclined slightly, about one-half of a degree down by the head, and the inrush of the water controlled by manipulation of the valves. All this is done in the short period of from one to two minutes. Seven boats have been known to average about two minutes for this evolution during maneuvers. If the control of the boat is difficult, ballast is taken on until the boat "handles" correctly; then trimming is stopped.

The duties of each member of the crew are clearly defined. At least two and sometimes all the men are thoroughly familiar with each station, so that when an order is given it is intelligently and promptly carried out. All orders are repeated. When an order has been executed the fact is reported to the commanding officer. In submerging a boat all precautions are observed before flooding the main tanks, and the fact that they have been followed is reported to the commanding officer or his assistant by the men at the various stations.

Submerging a submarine is distinctly a one-man job. The commanding officer must be thoroughly conversant with all the details of the actual submerging of the boat and he must at all times be thoroughly informed as to existing conditions in the boat. None of the important features can be delegated to anyone else, as each condition or state of affairs has a distinct relation to every other condition. The second officer assigned to the submarine generally assists the commanding officer where possible, and endeavors to acquire all the information possible on the details of the operation in order that he may take charge in case of any emergency.

Navigating a submarine on the surface is just like navigat-

ing any ship of the same size except that the displacement for amount exposed is greater. Care must be taken in making landings, as the headway is not so easily checked due to the available power for this purpose being less per ton generally than in ordinary vessels. Navigating submerged, that is, totally submerged, is just like navigating a ship in a fog on the surface, except that when the depth is great enough there are no passing steamers and fishing craft with sleepy crews to cause worry. Navigation submerged may be compared with the flight of an aeroplane in all respects except speed and the density of the substance through which passage is made. There are, of course, no "holes" in the water as there are in the air, but there are cross currents, strata of different density and other conditions to consider. A submarine uses the diving rudders for submerging or rising in much the same manner that a hydroplane or aeroplane uses the horizontal rudders to obtain the necessary angles for rising or gliding.

Those who have observed an aeroplane in flight have, no doubt, noticed the rolling of the machine. This is apparent in a puffy wind. In a submarine running submerged we have the same condition if the water on the surface is rough. A submarine rolls a slight amount at a depth of 50 feet if the surface is *very* rough, but practically all motion is lost at a depth of 75 feet. During the recent maneuvers in the English Channel, however, commanding officers of submarines reported that it was difficult to find a depth at which there was no motion. This is readily understood, as the waters in that vicinity are somewhat shallow and the ground swells can be felt quite near the bottom.

Depth is maintained by the use of the diving rudders. Down rudder sends the stern up and the bow down. As soon as the boat is inclined downward the plane of the deck gives a large surface for the water to act on, and the power delivered to the propellers may be resolved into two components, the horizontal, driving the boat ahead, and the vertical, driving the boat downward. The amount of the vertical component of the power delivered is controlled entirely by the diving rudder.

The hull is so built that normally there is a tendency for the boat to rise, brought about by the angle of the hull and the unbalanced pressures caused by inequalities in the areas of the surfaces on the bottom of the boat and on the top of the boat. This tendency must be counteracted by the trimming of the vessel and the action of the diving rudders. In some designs of submarines there are hydroplanes, forward diving rudders, and other devices to perform the same work, the difference being in diving at an angle or on an even keel.

The steering of the vessel in azimuth is done the same submerged as on the surface, using either hand or electric power. The magnetic compasses for submerged running are mounted above the hull in a composition hood and so arranged that the face of the card is reflected into the hull where the helmsman can readily see it. In recent years the gyroscopic compass has been perfected and has been installed in most of our submarines. This compass is operated entirely by electricity and the force of gravitation, both of which can be made independent of the magnetic influences of the earth. The gyroscopic compass must be adjusted for the different speeds and courses, but except for this it is automatic in its action and points in a true meridian with very little or no variation. The installation of gyroscopic compasses renders the tactical problems far simpler, as it is possible with this compass to steer the vessel within narrower limits than was possible with the magnetic compasses, owing to the fact that the action of the gyroscope is instantaneous, while the magnetic compass is very sluggish.

Control of the submarine is affected by speed. The effect of rudder manipulation varies in direct proportion to the speed of the boat. It is possible to handle a badly trimmed boat if sufficient speed is used.

It has been demonstrated that there are quite marked differences in density of water at different points and that this variation in density has a very appreciable effect on the trim and handling of a submarine. An account of an actual experience of certain United States submarines in which variation of density of water necessitated a difference of as much

as 150 to 200 pounds of ballast per mile will be found in an article entitled Notes on Submarine Repairs and Operations by the author of this paper in the JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS for August, 1914.

The periscopes are the eyes of the submarine. There are many styles of periscopes—monocular, binocular, stationary, revolving, rotating eye piece, stationary eye piece and "walk around." There are also devices known as omniscopes in which the entire horizon is reflected on a ground glass and any part can be separated from the rest by means of screen sectors. The periscopes are long tubes with lenses in the top and bottom and so fitted with prisms that the rays of light are paralleled. The general impression one gets is that of looking through a telescope.

The periscope that is in most general use in the United States Navy is the walk-around type with normal and magnifying eye pieces. Each boat is required to have at least two periscopes, one for the commanding officer and one for the helmsman or the second officer. In the recent boats, fitted with gyroscopic compasses, the helmsman takes his station at the compass while the lookout is at the periscope from which he directs the helmsman. The commanding officer usually stations himself at the after periscope and directs the movements of the vessel. Should he leave the periscope for any purpose, such as to consult chart or to get signals, some one always takes his place, so that there is never a time that both periscopes are not manned. When totally submerged, however, no one is needed at the periscopes for it is impossible to see more than a few feet through the water.

Signaling while submerged is a subject of much interest. In the early days of submarine navigation signaling under water was done in a most crude manner for which the hull of a vessel was found to be peculiarly adapted. Sending was accomplished by tapping on a rivet with a hammer and receiving by holding the forehead to a frame of the boat. For several years inventors have investigated and experimented extensively with the result that there are now in practical use in

submarine signaling the submarine bell, the Fessenden oscillator and the vibrating wire, by which it is possible to signal effectively at distances greater than five miles under favorable conditions. All these systems set up vibrations in the water which are detected by microphones and heard through the ordinary telephone receiver.

Inventors are now endeavoring to devise means for increasing the speed of transmission.

Each submarine is equipped with one bell, some with two, and on every submerged run, in peaceful maneuvers, these bells are rung so that it is possible for those in one boat to know the approximate location of other boats. The fact that no signals were received from the *F-4* has caused a belief that the vessel was so damaged that the crew remained alive for but a very short time after the accident which sank her occurred.

A phenomenon is the submarine "echo." The sound waves strike an obstruction, are reflected, and are picked up at the source. This could be used to determine approximately distances from obstructions, dangers, etc., but is not recommended as the possibilities of error are many.

On the surface submarines make use of the ordinary means of signaling—wig-wag, hand-semaphore shapes, flags and radio. The last is the best, but is not developed to that degree where it is absolutely reliable in submarines except under ideal conditions. However, it has proved its worth. Shape signals must necessarily be small owing to lack of space to handle them. Flag signals are inadequate for the same reason, but are carried for use in port. At night, light and pistol signals are used. Either of these systems is effective. A means of signaling which can be used very effectively both in the day time and at night is the flashing of the searchlight. While it seems rather unusual to employ a light for signaling in the day time, it can be used very effectively, and occasions have arisen when signaling with a submarine at a distance could be carried on in no other way.

The equipment of a submarine for the comfort of officers and crew is not very extensive. Officers generally sleep on

cots and the crew in hammocks. Ice boxes are installed on the later boats, enabling them to carry fresh meats and vegetables. Meals on these vessels are, therefore, more satisfactory than on the earlier boats. In addition to the fresh provisions carried, each vessel has a dry-food supply constituting an emergency ration sufficient to last 5 days. Fresh water is carried in tanks, and it is necessary that the crew be put on an allowance when at sea and that bathing be done over the side.

The submarine of the present day can operate at its maximum speed submerged for about one hour. At about one-third of this maximum speed she can operate practically twenty-four hours.

The offensive weapon of the submarine is the torpedo, or if necessity demands, the vessel itself can be used as a ram. The effect of the submarine as a ram may be illustrated by the fact that in a collision between a United States vessel and its tender several years ago, the tender was so badly damaged as to require beaching in order to save her.

The principal offensive weapon is, however, the torpedo. Ramming would be resorted to only in an exceptional case. The torpedo is an intricate mechanism built along the lines of the submarine itself, but automatic in its action after leaving the tube from which it is fired. The torpedo is divided into four main parts—the head, air flask, afterbody and tail. The head contains the explosive and the mechanisms to fire it. These heads are carried on the torpedoes only in war time. In time of peace exercise heads containing water are used. Abaft the head is the air flask. This contains the compressed air for the propelling machinery. In the next division of the torpedo is found the depth-controlling mechanism, the air superheater, the main propelling engines, the steering engines and gyroscope, and the shafting to the propellers, as well as the rods to the rudders. The tail inside contains the gear wheels for the propellers; on the outside are the supporting parts for the propellers, the diving rudders and the vertical rudders. The early torpedoes had an effective range of about

500 yards. The most modern product has an effective range of more than 10,000 yards. The development of the torpedo has been as amazing as the development of the submarine.

The operation of a torpedo is very similar to that of a submarine, except that while the submarine is operated by the crew in it, the torpedo is operated entirely by the automatic mechanism it contains.

The primary object of a submarine is to fire torpedoes, and all other considerations must be subordinated to this as far as the tactical value of the vessel is concerned, but not to such an extent as to lose sight of the fact that the torpedoes cannot be fired at a target unless the vessel arrives on the scene of action. The efforts of the crew are aimed towards one thing, sinking an enemy, and the placing of the boat in position to accomplish this end calls for the coöperation of all the departments in the boat.

An attack on a hostile vessel can be made in so many ways, and would have to be made under so many different conditions, that it is impossible to lay down any hard and fast method of procedure. The best time to attack is at dawn or twilight, or at a time when there is enough sea running to make the detection of the presence of the boat most difficult. Submarines have been known to approach within five hundred yards of a battleship in a choppy sea and in a fog to come up under the stern before being discovered, notwithstanding the fact that lookouts were stationed on the battleships to watch for the submarines. On smooth days the wake of a periscope may be picked up at a distance not greater than 5,000 yards. A submarine cannot, however, be distinguished at much more than five miles distant, at which point the boat would usually submerge and prepare to attack.

The best situation for an attacking submarine is forward of the enemy's beam and approaching her. From such a position it is possible to get within range most easily and lie in wait until the enemy crosses the path of the submarine. In such a case it is possible for the submarine to pick the time for firing with some degree of assurance that the torpedo will reach the

enemy and destroy her. Should the enemy be very much faster that the submarine, tactics become a case of hare and hound, with the advantage in favor of the enemy, as the speed of a submarine is limited.

The subject of safety in the submarine has received much attention and many devices have been invented designed to obviate the dangers said to attend the operation of this type of vessel. Each designer and builder has his own method of escape for the crew in case of accident.

It is believed that the one most effective and most nearly positive means of safety is eternal vigilance. In reviewing all the accidents that have occurred in the last twenty years, hardly any are found that cannot be attributed to some one thing that could have been prevented had the commanding officer been able to correct the fault. Submarines should be so operated, and it can be done, that all except remote possibilities are provided for.

In the United States Navy officers and men on submarine duty are given a thorough course of instruction. The men are required to be thoroughly conversant with all the parts of the submarine before they are qualified to receive the extra compensation allowed for submerged runs. The officers under instructions are trained by experienced officers and are not allowed to operate alone until they have shown that they are capable of doing so. This system has worked most successfully, the recent accident that caused the loss of the *F-4* being the first serious accident which has befallen a submarine of our Navy. The training goes on continually and all hands are impressed with the fact that submarine operation is most important duty.

It is true that accidents can and will happen, but conditions which make for catastrophe generally do not arise so rapidly that some means cannot be taken to get the boat up and the crew to a place of comparative safety before the actual accident takes place, provided, of course, that the vessel does not get beyond her safe depth. In case of accident at a depth exceeding 100 feet, the chances of getting out are remote. Even if

a man escaped, he would have little chance of getting to the surface alive owing to the sudden changes of pressure.

The mental condition of officers and crew in the event of accident should receive most careful consideration in the design of any means for escape, and the aim should be to produce a device that can be operated with certainty by even the panic-stricken.

As submarine design improves, however, dangers of operation are becoming fewer. Today the submarine possesses no more inherently dangerous features than are possessed by any ship. In fact, they are safer to cruise in than a surface craft and for the same tonnage are far more comfortable in a heavy sea.

The personnel of a submarine depend for air for breathing purposes while submerged on the free air in the boat at the time of submerging and the compressed air carried in storage flasks, which is used in freeing ballast tanks of water as well as for breathing. In the average submarine in commission today, the air contained in the boat at the time of submerging is sufficient to last officers and crew, numbering 18 men, for a period of from 9 to 12 hours. The air carried in the storage flasks is about sufficient to replenish the entire volume of air in the boat twice at atmospheric pressure, provided it is used for no other purpose. The maximum time during which all the air available can be breathed without serious effect is, therefore, from 30 to 36 hours. In computing this time, the safe amount of CO_2 that should be allowed to accumulate in the air at any time is taken at 2 per cent. Men vary, however, in their ability to withstand the effects of CO_2, the average man being able to withstand about 2½ per cent., while an exceptionally strong man can withstand as much as 5 per cent. Therefore, the time during which life can be sustained by the air in a given boat will depend somewhat upon the powers of resistance to the effects of CO_2 on the part of the personnel.

The air is maintained in condition for breathing by two methods: first, by slowing bleeding from the main air supply into the boat and pumping air out very slowly; and, second,

by allowing the air to become foul and then pumping a part of it out of the boat and replenishing it from the air flasks, at the same time maintaining a normal atmospheric pressure in the boat. The former method has been found more economical of air. The air in the boat is kept in circulation, as it has been demonstrated by experiment that air in circulation may be charged with a much greater percentage of CO_2 than still air without evil effect on the person breathing it.

THE FUTURE OF THE SUBMARINE.

The future of the submarine opens a large field for speculation. It is believed that the size of the boats, the general design, the motive power and the principal characteristics will not change materially in the next few years, except in the case of the sea-going submarine which is now being built and which departs greatly from all previous designs.

As far as can now be foreseen, the future power of submarines may be expected to be the oil engine or some modification of it. However, engineering fields have not been sounded to their depths in the determination of adequate power for these vessels, and it is believed that the future may see some development and improvement, and perhaps a material departure from any methods of propulsion heretofore adopted or proposed.

With several flotillas of the sea-going type of submarines above to cruise with the fleet, they might one day be found in the line of battle with the destroyers. Some authorities claim that this type of submarine will eventually supplant the torpedo-boat destroyer. This is considered most improbable, however, as, owing to their shape, high speeds call for prohibitive powers, and, with the methods of propulsion now in use or contemplated, it is not reasonable to expect a speed greater than 25 knots on the surface or 15 knots submerged.

NOTES.

THE BATTLE-CRUISER IN WAR.

By Archibald Hurd.

(*Author of "Command of the Sea," "Naval Efficiency," etc.*)

Every notable success of the British Navy in the present war has been due to the battle cruiser. By the employment of ships of this new type the enemy on three successive occasions has been surprised owing to the superior mobility of the British force, and then overwhelmed by superior weight of metal in combination with, at least, not inferior gunnery.

The triumph of the battle cruiser bears testimony not only to the foresight exhibited by the British Admiralty, but attests the competency with which they applied the historical principles of war to the new conditions which were already coming into view when the battle-cruiser type was evolved. Had the British Fleet not possessed ships of this class, the discomfiture of the enemy in the Bight of Heligoland in August last would have been impossible; Admiral Graf von Spee's squadron could not have been practically annihilated in the course of a few hours on December 8th; and on January 24th, when the enemy set out to bombard once more the East Coast of England, his retreat could have not been cut off, the *Blucher* sunk, and two other large units seriously damaged. But for the battle cruisers, the British Navy could have made little progress during the past seven months in weakening the enemy, and would not have had much to show as an offset against our losses inflicted by the Germans by the use of submarine and mine. Generally it may be said that the Germans owe practically everything to invisible attack, and the British to swift surprise, supported by heavy gun power. Commenting upon the Dogger Bank action, the First Lord of the Admiralty in his speech in the House of Commons, on February 15th, remarked that it vindicated, "so far as it went," theories of design, and particularly of big-gun armament, always associated with Lord Fisher. The range of the British guns was found to exceed that of the Germans. Although the German shell is a most formidable instrument of destruction, the bursting, smashing power of the heavier British projectile is decidedly greater and—this the great thing—our shooting is at least as good as theirs.

What manner of ship is the battle cruiser which has contributed so greatly to the promotion of the British cause on the sea. Mr. Kurt Orbanowski, a naval constructor of Polish birth, has recently given an interesting definition of this type of man-of-war. Mr. Orbanowski was formerly with the Hamburg Shipbuilding Yard, and was responsible for the design and construction of the armored cruisers *Scharnhorst* and *Gneisenau*, and, later on, of the battle cruisers, *Moltke* and *Von der Tann;* subsequently when the Russians established their large shipyard on the Gulf of Finland, he was placed in charge of it, and designed the battle cruisers of the *Borodino* class of 32,000 tons, a nominal speed of 27 knots, an armored belt of 13 inches, and an armament of twelve 14-inch guns in association with twenty-one 5.1-inch guns.

Dealing with the battle-cruiser type, Mr. Orbanowski recently declared that without a sufficient number of this type a battle fleet remains a torso,

apt to be attacked at a tactically disadvantageous position by a powerful adequately equipped enemy. "A battle-cruiser squadron as a fast division, or fast-division wing, of the battle line, can, because of superior speed, quickly accumulate power at the enemy's weakest point, attacking head or rear unexpectedly, or repulsing similar attacks of the opponents. Their strategical value is without doubt. They can cut off and repulse the enemy's scouting forces (cruisers) and throw a complete veil over the movements of their own battle fleet, especially on the high seas, where the use of air craft is still very limited." Their speed, he contended, enables them to rush to points of a coast menaced by an enemy's raids, which is especially valuable for a country with such far-extended and double-sided coasts as the United States, with many great industrial towns in reach of attack. "Their powerful armament holds the enemy in check until battleships arrive. The essential qualities of a battle cruiser are heavy guns of the same, or nearly the same, caliber as those of battleships, slightly less in number; slightly less thickness of armor; and high speed." According to the investigations of naval experts with whom he has been in touch, Mr. Orbanowski stated, their speed should be twenty-five or thirty per cent.—or at least 5 knots—more than that of the battleships. As battleship speed, up to the latest types, was 21 knots, battle cruisers made 26 to 28 knots. The increased speed of new battleships to 22 or 23 knots necessitated an increased speed of 29 to 30 knots for battle cruisers. "Higher speeds than these at the present stage of marine engineering, aside from any unexpected developments, cannot, in his opinion, be counted upon.* "Therefore we meet already for solution the problem of the intermediate type of battleship, using oil fuel only, such as the British *Queen Elizabeth*,† or the Italian type, with 25 knots speed."

These considerations led the two leading naval Powers now engaged in war, Britain and Germany, to develop the battle cruiser side by side with the battleship of the Dreadnaught type. Japan, and much later, Russia, followed the example. Smaller nations, such as Italy and Argentine, adopted the intermediate-type of speedy battleship (23 to 25 knots). The British battle cruisers number about thirty per cent. of the number of the modern battleships, even if we take into account the intermediates of the Elizabeth type (25 knots). There are 4 Invincibles, 4 Lions, 2 New Zealand and Australia, altogether 10; against 23 Dreadnaughts (21-23 knots) and 5 intermediate Dreadnaughts (25 knots)—altogether 28; plus 1 Queen Elizabeth (27,000 tons) and Royal Sovereigns (25,300 tons) now building.

Germany has 7 battle cruisers, 1 Von der Tann, 2 Goeben, 1 Seydlitz, 1 Derfflinger, 2 Luetzow; against 17 Dreadnaught battleships, 4 Nassaus, 4 Helgolands, 7 Koenigs, 4 Kaisers; besides 1 battle cruiser and 1 Dreadnaught building, *i. e.*, nearly forty per cent. of the number of their battleships. Russia's naval program proposes a similar proportion, *i. e.*, battle cruisers one-third of their battleships. She has already four big units of 33,000 tons (29 knots) on the stocks. It is a notable fact that down to the outbreak of war neither the United States, France, Austria-Hungary, or either of the South American or Latin American Powers had laid down a single battle cruiser. Why? The matter is one of some interest in view of the leading events of the present war on the seas.

The success with which the British Fleet has been able to employ these ships throws into prominence the criticism with which the new type was assailed, criticism, moreover, which apparently was regarded by a large number of British and foreign naval officers as conclusive. When the first three battle cruisers of the Invincible type were laid down and then for two years in succession no provision was made for other units of the

*In this statement Mr. Orbanowski will probably be shown to be in error.
†The British Royal Sovereign battleships are also to have liquid fuel only.

same character, an anonymous writer in the "Naval Annual" declared that it was apparent that the British Admiralty were "not going to build any more huge armored cruisers," and it was added that "we must hit upon some plan of employing those which we have already." Admiral Sir Reginald Custance was equally decisive in expressing his opinion of the future of the new type of man-of-war. He claimed that "by argument the class have been killed" and it only remained "to inter them decently away from the public gaze." Admiral Sir Cyprian Bridge, in the spring which witnessed the appearance of the first British battle cruisers, wrote an interesting article on the lessons of the war in the Far East. He declared:

"A ship of war is intended principally to fight and not to run away. We should therefore be careful not to give to any other element undue predominance over the element of offensive power in the design of a ship meant to be capable of destroying or defeating her antagonist. In ships for fighting general actions—that is ships for fighting in combination with consorts, the element of offensive power in any individual ship should bear the proper relation to the aggregate of that power in the whole group. Suitable dispersion should be given to the instruments of offensive power, and allowance should be made for suitable concentration of their effect. For certain classes of vessels, which usually will be of small size, very high speed, greater than that of an antagonist, if possible, should be provided; but it must be understood that these vessels can play only a special and restricted part in war."

It should be added that this statement was written before anything was known of the design of the Invincible class. Sir Cyprian Bridge confined himself to a statement of views based upon his interpretation of the action which had recently been fought in the Far East.

The most strenuous opponent of the new type was Sir Reginald Custance. By the time this officer wrote, a good deal was known of the new ship, and he held that he was in possession of sufficient facts to justify criticism. He contended in "Naval Policy; A Plea for the Study of War," in the first place that "neither in practice nor in theory has it ever been proved that superior speed gives any tactical advantage, unless it be thought an advantage to be able to run away.

"* * * Speed is not a weapon and does not give protection, except in running away. The aim should therefore be to endow a fleet not with superior speed or protection, but with superior offensive power, *i. e.*, gunpower." He denounced the Board of Admiralty which was responsible for the new design, claiming that it consisted of men "who had no practical war experience and, not having studied history, are not familiar with its leading principles." He added that, "Our belief is, that future wars will show the necessity for a large number of capital ships moderate in size, supported by a small percentage exceptional in power; that as these ships are to act together their speed should be the same, and should be that best suited to the more numerous class; that the exceptional ships should excel in fighting power, and not in speed."

In 1912 Sir Reginald Custance published a book entitled "The Ship of the Line in Battle," and in the course of his discussion of the co-related problems of gun-power, armor and speed, he first of all dealt with the armored cruiser and contended that "the armored-cruiser policy was not based originally on true conceptions of war." It involved, he urged, not only a faulty strategy, but mistaken ideals, both of which reacted on ship design. "The essential feature of the armored-cruiser design was a sacrifice of fighting power to mobility in the ship of the line. If it is admitted that the policy was based on faulty strategy and tactics, that sacrifice cannot have been justified. The proportion of fighting power allotted to armor in the ship of the line may have been excessive, but what justified the sacrifice of guns? What are the reasons for it now?

Are they tactical? Is it quite certain that in war games and tactical exercises during peace undue importance is not attached to supposed tactical advantages, and that actual fighting is not relegated to a secondary place in the mind? Are the conclusions based on true premises? In time of peace make-belief under mistaken ideals may be practised with impunity, but in war there is no make-belief: Then realities will have to be faced, as by the Russians at Ulsan."

Towards the end of this interesting volume, written by an officer with considerable sea experience, a student of naval warfare and a former Director of Naval Intelligence at the Admiralty, certain specific principles were laid down which in the light of events may be profitably recalled. Dealing with the battle of Shushima, Sir Reginald Custance observed that the facts "confirm previous war experience that the danger to the flotation and stability is not great. Is it worth while to divert from the guns the great weight required to give effective armor protection to the waterline, when the chances are that the battle will culminate before it is hit? Will it not suffice to make sure that the magazines are safe from direct blows and, for the rest, to trust to watertight sub-division, to armor only so far as it may limit the size of such holes as may be made, and above all to gunfire to beat down that of the enemy? Is it not more important to disarm the enemy than to sink him? Are not the protection of your own waterline and the perforation of that of the enemy secondary considerations in settling the armor and guns to be carried?"

Sir Reginald Custance then dealt in general terms with what has come to be known as the all-big gun principle:

"Now the main object in battle is to make the enemy believe that he is beaten. The most effective way to do this is to disable his *personnel* and silence his guns. The above results (that is the results of the battle of Tsushima) seem to indicate that the smaller gun is by no means to be neglected as an instrument for this purpose. The effect produced depends, not only on the size of the projectile but on the place where it hits. A small shell on the right spot is more effective than a large shell in the wrong one, but to hit the right spot is difficult. Hence, in determining the armament of a ship, a careful balance must be maintained between the numbers and sizes of the guns carried. Again, the facts show that it is misleading to compare the gun power of ships by the total weights of their respective broadsides. To do so is to assume that on the average an 850-pound 12-inch shell will damage the fighting efficiency of the ship as much as will eight 100-pound 6-inch. Such an assumption seems not to be true. When the guns in ships of the line were all about the same size, the method was legitimate, but it is believed to be entirely misleading at the present time when they differ so much, some being, perhaps, unnecessarily large and others too small for the work to be done. Are not the numbers and sizes of the guns carried the best and only safe standard of comparison?

"Thus we see that whether we consider the difficulty of hitting or the comparative effect produced by shells of different calibers, there are grave doubts whether batteries of comparatively few large guns form the most effective armaments. Any reduction in the size, or change in the disposition, of the guns will at once react on the size of the ship. All vertical armor in existing ships is now perforable by high-explosive shell, and does not seem to return sufficient value for the weight it absorbs and diverts from guns, except against fragments of bursting shells. Moreover, ships can be beaten without perforating the armor—*e. g.*, *Orel*. The smallest gun required to maintain the ascendency over the armor must be carried in sufficient numbers, but should not these be supplemented by others of smaller size to increase the volume of fire and to provide a larger margin for misses and failures? Ascendency over the armor means that victory can be won in spite of it, that is to say, by

perforating it if spread out, and by overwhelming the ships and crew if concentrated."

No apology is necessary for quoting at considerable length the varied and instructed criticism with which the battle-cruiser type was assailed, particularly as this line of criticism was apparently endorsed by the Navy Department of the United States. The battle cruiser was denounced as a ship faulty in design, the principal characteristic of which was ability to run away and inability, owing to the absence of small guns, "to make the enemy believe that he is beaten." It was, moreover, repeatedly affirmed that such vessels were really capital ships, and as such would be massed in fleets with the battleships; no admiral, it was contended, would have the courage to spare such units for detached service.

During the course of the war no action has occurred in which the battle squadrons of either the British or German Fleets have been engaged. On the other hand, the Germans have on three occasions employed a force of battle cruisers, the objective in each case being to bombard defenceless towns on the East coast of England. Why? it may be asked, should battle cruisers have been employed, since this type of ship is little faster than a number of the light cruisers under the German flag? In the first place, the big ship holds its speed better in a heavy sea; in the second place, the big ship carries guns which can overwhelm small craft. The intention of the Germans in each instance was to cross the North Sea during the hours of darkness. If British light cruisers or destroyers were encountered, the heavy gun could put them out of action; if submarines were met with, the high speed of the battle cruisers would in itself prove a protection against attack. The first two attempts to carry out this policy were successful in that the German ships reached the British coast, fired their guns and returned home without loss.

It was on the third occasion—on January 24th—that an exhibition was provided of the value of the bigger gun carried in the British ships and of their superior speed. The Germans employed on this occasion—the *Moltke,* credited in the "Naval Annual" with a trial speed of 28.4 knots; the *Seydlitz,* reputed to have obtained 29.2 knots; the *Derfflinger,* with a rate of steaming superior, it was believed, to that of the *Seydlitz;* and the *Blucher,* a contemporary ship to the British *Invincibles,* with a trial speed of 25.3 knots. Admiral Sir David Beatty's Squadron consisted of the following units:

Name of Ship.	Built at.	Engines Built by.	Speed.
Lion	Devonport	Vickers	28.5
Tiger	Clydebank	J. Brown	28
Princess Royal	Barrow	Vickers	28.5
New Zealand	Govan	Fairfield	25
Indomitable	Govan	Fairfield	27.3

As the speed of a squadron is that of its slowest ship, the German force was seriously handicapped owing to the presence of the *Blucher.* The immunity which had been enjoyed on the two previous occasions had apparently led the Germans to anticipate that on January 24th the raiding force would again not encounter any vessels of the British Fleet, with heavy guns, which could steam faster than the *Blucher.*

The German ships, when sighted, were approximately 14 miles east-south-east of the British force. Sir David Beatty steered south-east with a view to securing the lee position, and, if possible, cutting the enemy off from his base. The British admiral states that "speed was worked up to 28 and 29 knots, and the enemy were gradually being overhauled." At about 18,000 yards "slow and deliberate fire" was opened, and it is added, "we began to hit at a range of 17,000 yards." The *Tiger,* as was, perhaps, to have been expected, and the *Lion,* drew ahead of the

remainder of the squadron and, while themselves exposed to heavy fire, were able to punish the enemy severely. At last the German admiral decided to sacrifice the *Blucher*, and he returned to port at the highest speed, with two of his battle cruisers injured and heavily on fire, although apparently their stability was not seriously affected by any waterline hits. The *Lion*, on the other hand, was penetrated at the waterline by an 11-inch shell and, owing to her condition and the presence of the enemy's submarines, the action was broken off.

In view of the character of the two squadrons engaged, the result of the action is illuminating. If the German admiral, as was apparently his decision, determined not to stand by the *Blucher* and not to fight, he ought to have been able, on the trial speeds of his three battle cruisers, to return to port without being hit by a single British shell. On paper his three ships were at least as fast as, if not in one case faster, than the British vessels. Before the war occurred any German officer would have supported the contention that the three German ships could show their heels to the British vessels. In practice this proved not to be the case. Sir David Beatty's report goes to show that the *Lion* and *Tiger* gained rapidly on the three opposing battle cruisers and were thus able to bring them within effective gun fire and do them considerable damage. Owing to the principles in design for which Lord Fisher was responsible the British ships mounted thirty-two 13.5-inch guns and sixteen 12-inch guns. On the other hand the German squadron mounted eight 8.2-inch guns, twenty-nine 11-inch and eight 12-inch. In retreat the Germans were able to bring into action about the same number of weapons as the British ships could train upon them, but the latter possessed the more deadly weapons.

The action was a triumph of the all big-gun principle, but it was no less a triumph for the British engineering firms concerned in the construction of the vessels, and for the engineer officers and their staffs. If the British ships, in the emergency, had not realized their speed expectations, the action would have been inconclusive. The chase was a stern one between two opposing squadrons, with no great disparity in their paper speeds; the British engines and boilers in the hour of trial proved superior to the German engines and boilers. The engagement forms an interesting commentary upon the boastful statements in which German firms indulged during peace, and exhibits in the eyes of the world the absurdity of the allegation put forward by Krupps to the effect that the British wire-wound gun was a weak and unreliable weapon.

What other type of ship but the battle cruiser could have brought Admiral Graf von Spee's squadron to action off the Falkland Islands? The enemy force by superior speed and superior gunfire had sunk the weaker British cruisers, *Good Hope* and *Monmouth*, off Coronel, on November 1st. On December the 8th four of the five enemy ships, under the German Admiral, were sunk off the Falkland Islands. Between these two dates the British Admiralty had carried out a concentration movement unknown to the enemy. In the time available battleships could have steamed from a British home port to the Falkland Islands, and could have reached Port Stanley on December 7th. The Admiralty, however, were aware, on the one hand, that they possessed no battleships with a speed of more than about 21 knots, and that the German Admiral had under his flag the *Scharnhorst*, with a speed of 22.5, and the *Gniesenau*, which on her trials attained a speed of 23.8 knots. Associated with this squadron were three light cruisers, the *Leipzig* of 23 knots, the *Nurnberg* of 23.5 knots, and the *Dresden* of 27 knots. It must have been apparent, therefore, that battleships of 21 knots could not overhaul the German ships, and that something more speedy was necessary if that object was to be attained. Under the British flag there were light cruisers which could steam faster than the German flagship and her armored consort,

but no such vessel carried anything heavier than the 6-inch gun, while the two principal German ships were armed with sixteen 8.2-inch weapons.

In these circumstances the British Admiralty decided that if the German force was, first, to be overhauled and then to be defeated, the ideal type of vessel for the purpose was the battle cruiser. The naval authorities, therefore, despatched from England, under Vice-Admiral Sir Doveton Sturdee, the *Invincible* and the *Inflexible*, with a nominal speed of 26 knots and each mounting eight 12-inch and sixteen 4-inch guns, in association with armored belts of a maximum thickness of 7 inches—a thickness slightly in excess of that possessed by the two principal German vessels. The *Invincible* was completed at Elswick in 1909, being provided with Parson's turbines by Messrs. Humphrys and Tennant; the *Inflexible* was constructed at Clydebank, and was engined by Messrs. J. Brown and Co. Providing no serious breakdown occurred in the ships under the White Ensign, the result of an encounter with Graf von Spee's squadron was inevitable. In the event, British engines maintained their high reputation and British engineers and their staffs showed how efficiently they could be worked. The two large ships of the German squadron were sunk, and, owing to the foresight exhibited by the British Admiralty, two of the three light German cruisers were also destroyed—the *Dresden*, the swiftest of the trio, alone escaping, which shows that the enemy found superior speed in this case no mean asset. The success of this rounding-up movement was due to the orders which resulted in the British cruisers *Carnarvon, Cornwall, Kent, Glasgow* and *Bristol*, being concentrated on the Falkland Islands, together with the auxiliary ship, the *Macedonia*, for the decisive event. Neither of the three British armored cruisers was of recent construction, as the following statement shows:

Name of Ship.	Built at.	Engines built by.	Speed.	Completed.
Carnarvon	Beardmore	Humphrys.	23.3	1905.
Cornwall	Pembroke	Hawthorn.	23.68	1904.
Kent	Portsmouth	Hawthorn.	21.7	1903.

These three ships, together with the *Glasgow* and *Bristol* (built at Govan and Clydebank, and engined by Fairfield and Messrs. John Brown and Co., respectively), amply justified the confidence of their builders, in spite of the years which had elapsed since they were completed.

The story of the *Kent's* pursuit of the *Nurnberg* is a romance of marine engineering. The story has been told by Midshipman John Esmonde in a letter to his father, Sir Thomas Esmonde, M.P.

The *Kent*, a 21-knot cruiser, was ordered to chase the *Nurnberg*, a 25-knot ship and also a much more modern one than the *Kent*. She had only a few hundred tons of coal on board to catch the *Nurnberg* with. The old *Kent* set off, and they worked up to 22, more than she had ever done on trials. Then the word was passed up that there was hardly any coal left. "Well," said the Captain, "have a go at the boats." So they broke up all the boats and smeared them with oil and put them in the furnace. Then in went all the arm chairs from the wardrooms, and then the chests from the officers' cabins. They next burnt the ladders and all—every bit of wood was sent to the stokehold. The result was that the *Kent's* speed became 24 knots, and she caught the *Nurnberg*, and after a stiff fight, in which several men were killed, the *Nurnberg* was sunk.

The performance of the *Kent* on this occasion will not soon be forgotten and, when it is recalled, tribute will be paid not only to the firm responsible for the engines, but to Engineer-Commander George E. Andrew, Engineer Lieut.-Commander Alfred E. E. Rayner, Engineer Lieut. (retired) Victor O. Foreman, and the engine-room staff, who must have put forth almost superhuman efforts to achieve so notable a success.

The battle cruiser has come to stay. It represents an effective compromise in warship design, particularly for a rich nation, with world-wide, home, commercial and vast territorial interests, determined to exercise command of the sea in time of war. High speed enables a concentration to be carried out swiftly and confers no mean tactical advantages when face to face with the enemy, while the association of high speed with an armament not inferior in caliber to that of a battleship contributes, as experience has shown, to decisive results, at a slight cost in casualties. The battle cruiser is "not dead"; there is certainly no intention "to inter it away from the public gaze." The tendency of development of this type of ship is revealed in the vessels of the *Kongo* and *Queen Elizabeth* classes, which are now passing, or have passed, under the Japanese and British flags. The *Kongo,* like the British ships of the *Queen Elizabeth* type, mount eight big guns only, but, whereas the Japanese ship carries the 14-inch weapon of the Vickers type, the British ships have the new 15-inch gun. Everything suggests that the future armored ship will consist of a compromise in which speed and gunpower will be further emphasized.—"Cassier's Engineering Monthly."

SOME TESTS MADE AT THE ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

FROM REPORTS PREPARED BY D. J. MCADAM, CHEMIST.

BOILER TUBES.

1. Thirteen boiler tubes were selected at random from about 200 tubes that had to be cut out of boiler drums intended for the *Jacob Jones* and *Wainwright.* The ends of the tubes had split when they were expanded and belled; about 200 out of 9,300 of the tubes proved defective.

2. The tubes were manufactured by the Pittsburgh Steel Products Company. Since the manufacturers could not trace the material back to the heat numbers of the steel, the results of their chemical analysis could not be obtained.

3. The purpose of the test was to make a chemical and metallographic examination of the samples received in order to determine the quality of the metal.

4. The tubes, after being photographed, were each cut longitudinally into two pieces for metallographic examination. Samples for chemical analysis were also taken. The arrangement of the metallographic specimens, numbers from 1 to 26, is as shown in Plate I. After the metallographic examination, ductility tests were made on pieces in their original condition and on pieces heat treated at this station.

5. The results of the chemical analysis were as follows:

	No. of Determinations.	*Average.*
Carbon	2	0.166
Sulphur	1	0.029
Phosphorus	2	0.042
Manganese	1	0.392
Silicon	1	0.037

The results of the metallographic examination are illustrated by the photomicrographs on cards 1 to 18. In all these photomicrographs, the white areas represent ferrite and the dark areas represent pearlite.

Card 1 shows the outer surface of specimen 1 near the crack shown on

Plate I. The magnification is 100. In this region the ferrite grains are elongated; the pearlite also occurs in streaks extending lengthwise with the tube. The quantity of pearlite varies greatly in different regions on the surface of the tube.

Card 2 shows the outer surface of specimen 2 near the crack shown on Plate I. The magnification is 500. The pearlite, which is less in quantity than in the area shown on card 1, occurs in segregated regions surrounded by relatively large regions of pure ferrite. In such a structure, the brittle high-carbon regions would behave to a certain extent like slag or other foreign particles. The pearlite in this condition has very little strengthening effect on the metal.

Card 3 shows the outer surface of specimen 4 at a magnification of 100. In this specimen the pearlite is unevenly distributed and occurs in irregular masses extending lengthwise with the tube.

Card 4 shows a longitudinal section of specimen 5 at a magnification of 100. The ferrite grains are distorted and the pearlite is very unevenly distributed.

Cards 5 and 6 show the outer surface of specimen 8 at magnifications of 45 and 100, respectively. The ferrite grains are distorted and the pearlite occurs in streaks extending lengthwise with the tube. A broad streak of nearly pure ferrite containing slag particles is seen in the center of the photomicrographs.

Card 7 shows a longitudinal section of specimen 10 at a magnification of 100. On this surface the distribution of ferrite and pearlite is fairly uniform and there is no distortion of the ferrite.

Card 8 shows a transverse section of specimen 11 at a magnification of 45. Alternate layers of ferrite and pearlite extend around this section of tubing. This segregated structure is due to cold working followed by insufficient annealing.

Cards 9 and 10 show two views of specimen 13. Card 9 shows a longitudinal section at a magnification of 45. Pearlite streaks and banks of nearly pure ferrite are seen in this photomicrograph. Card 10 shows the outer surface of the same specimen at a magnification of 100. The area photographed is in line with a crack shown on Plate I. Much segregation of ferrite and pearlite is found in this specimen; regions of more than average carbon content are surrounded by areas of nearly pure ferrite. In the streaks of ferrite are a few slag particles.

Card 11 shows the outer surface of specimen 16 at a magnification of 100. Though the structure of this specimen is better than that of some of the other specimens examined, there is considerable segregation of pearlite. The pearlite is found in small masses unevenly distributed and surrounded by large areas of nearly pure ferrite.

Card 12 shows the outer surface of specimen 17 at a magnification of 100. The ferrite grains are elongated in the direction in which the tube was drawn; the pearlite also occurs in elongated masses.

Card 13 shows the outer surface of specimen 20 at a magnification of 100. In this specimen there is some distortion of the ferrite grains, they are elongated in the direction in which the tube was drawn. The pearlite is small in amount and unevenly distributed.

Card 14 shows the outer surface of specimen 22 at a magnification of 100. The area here shown is near a crack in the metal. In this specimen there is much segregation of pearlite. Surrounding a region of nearly pure ferrite, seen in the central part of the photograph, are areas having excess of pearlite. A decided streak of pearlite is visible in the upper part of the photograph.

Card 15 shows a longitudinal section of specimen 23 at a magnification of 100. Elongation of the ferrite grains in the direction in which the tube was drawn is noticeable in this section. The pearlite also occurs in longitudinal streaks.

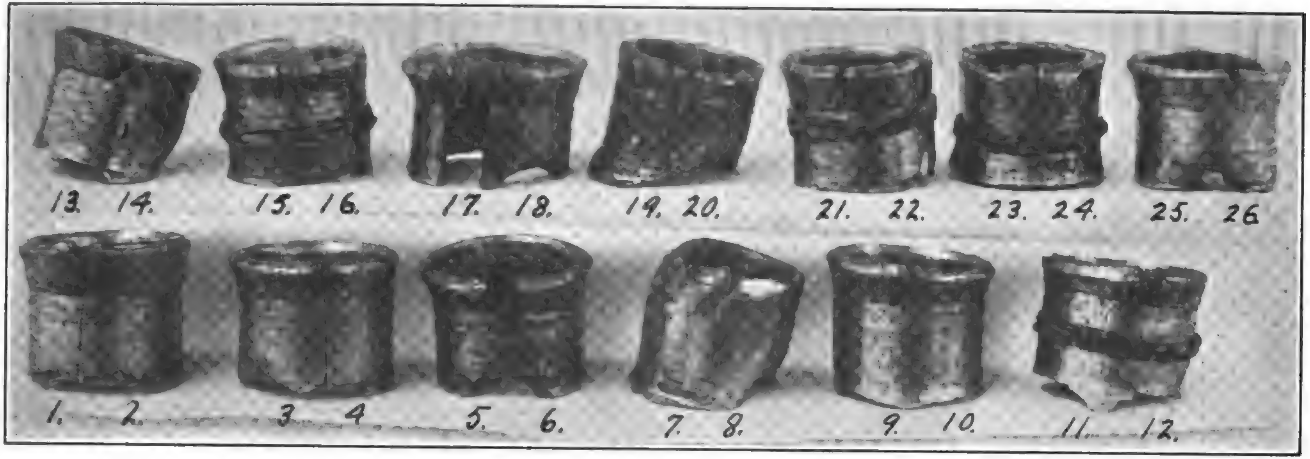


PLATE I.

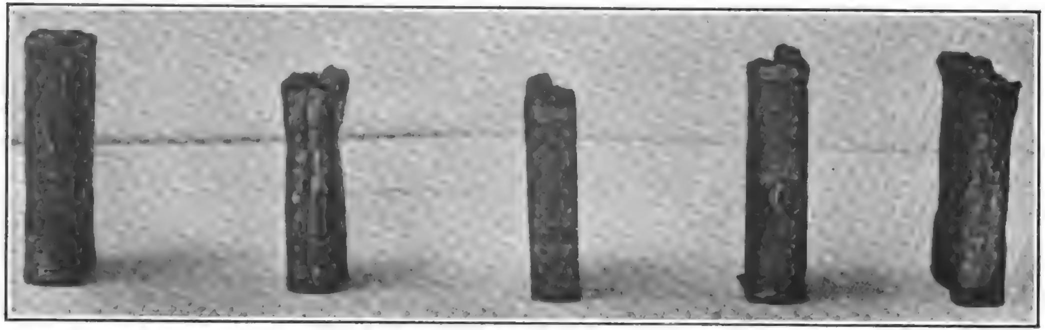
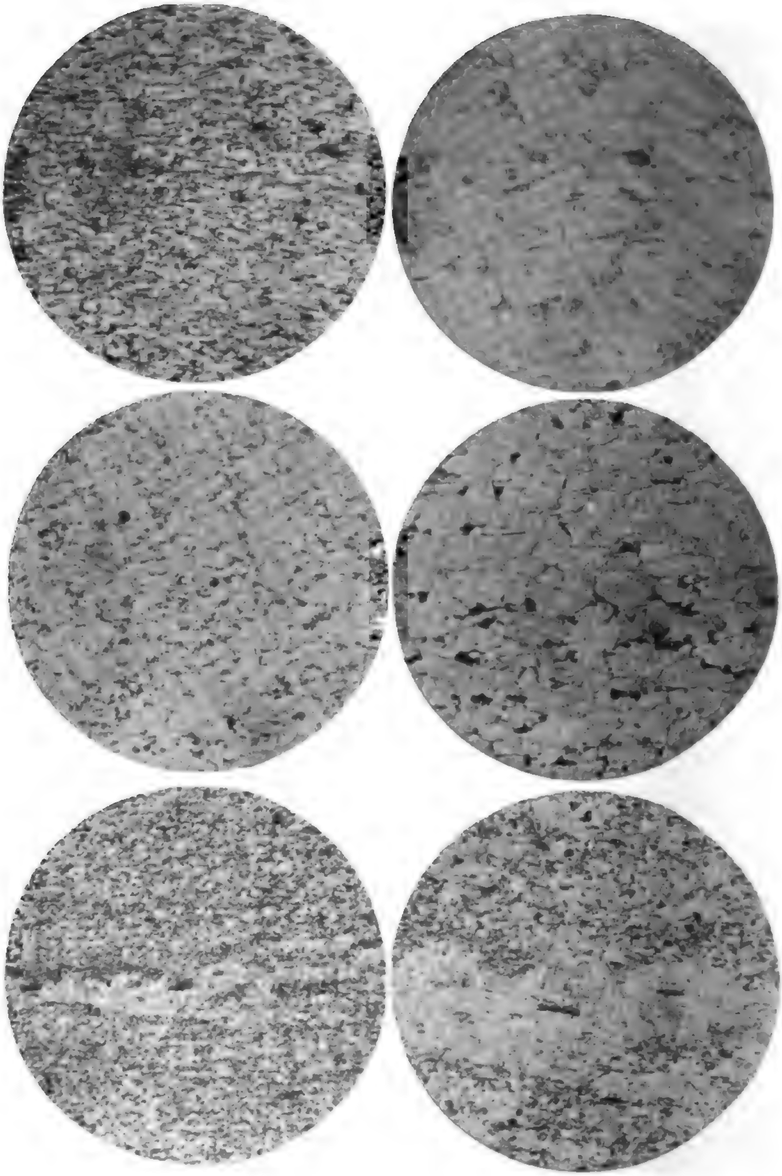
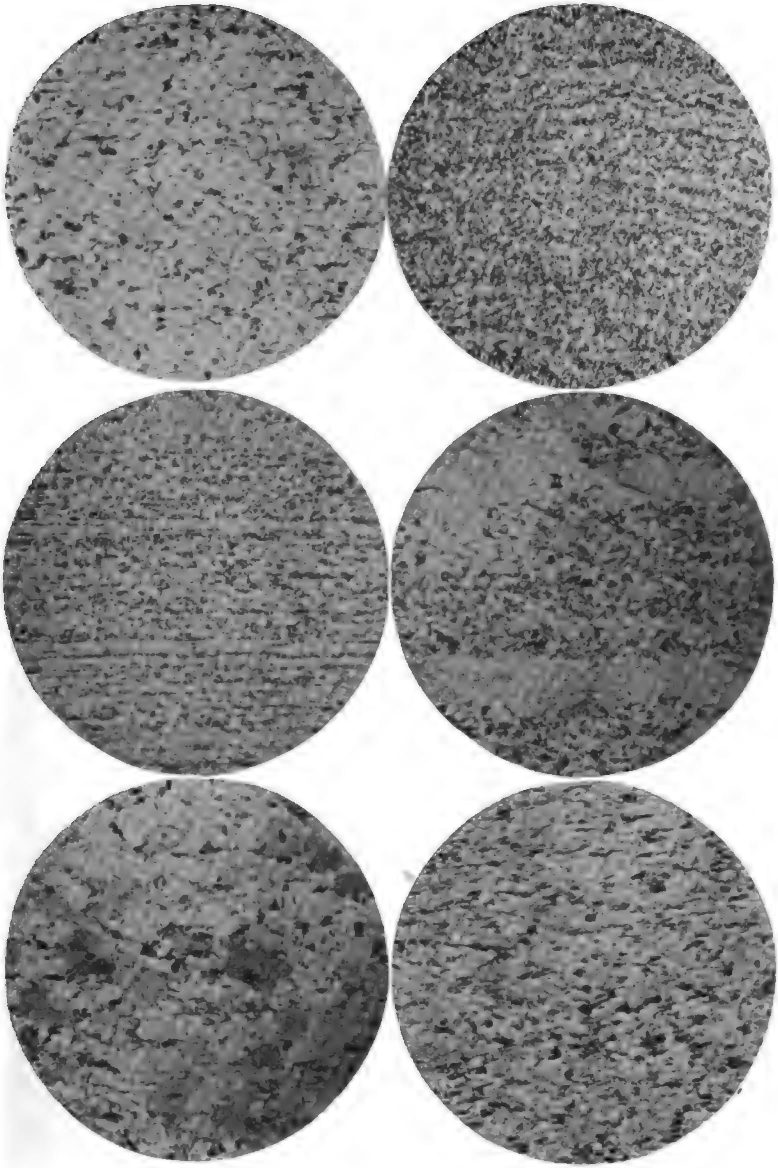


PLATE II.



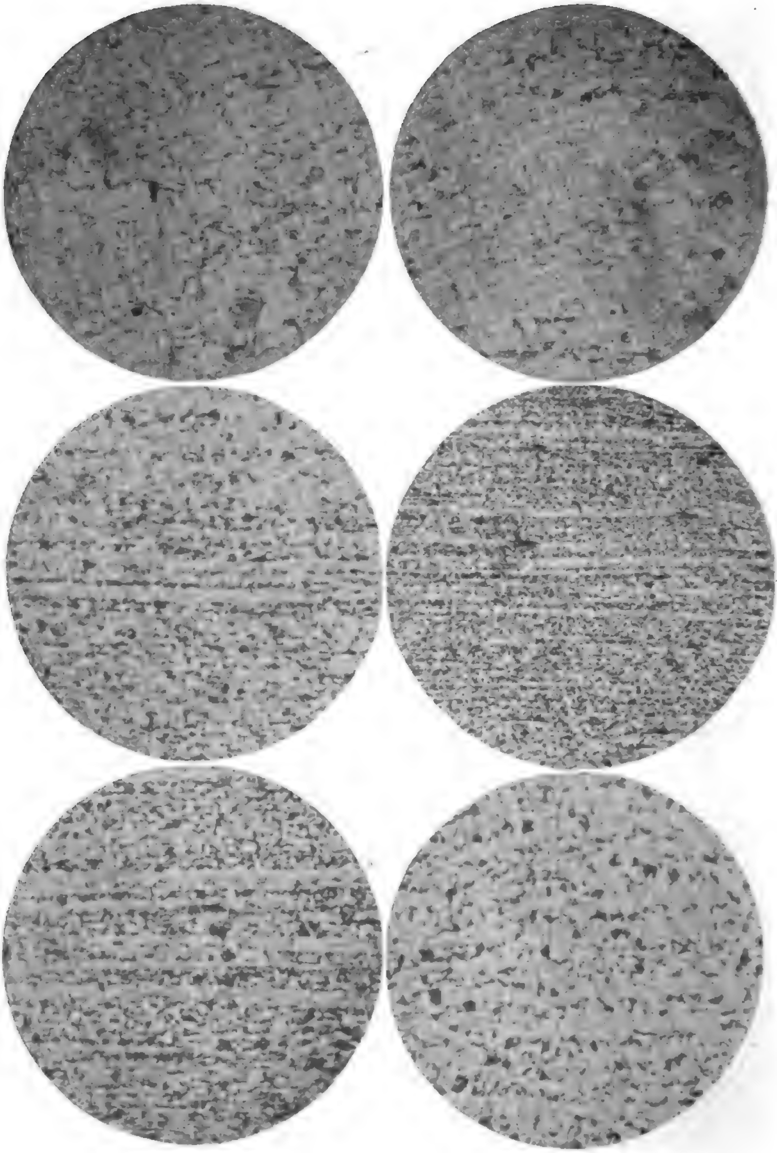
Card 1.
3.
5.

Card 2.
4.
6.



Card 7.
9.
11.

Card 8.
10.
12.



Card 13.
15.
17.

Card 14.
16.
18.

Cards 16 and 17 show a longitudinal section of specimen 25 at magnifications of 45 and 100, respectively. Streaked segregation of ferrite and pearlite is prominent in this specimen. Distortion of the ferrite grains is also noticeable.

Card 18 shows a transverse section of specimen 11 after heat treatment at the Station. It was heated at 1,600 degrees F. for two hours and cooled in the furnace. The original structure was as shown on card 8. Though some segregation still remains, the heat treatment has removed most of the effects of cold working. Pronounced streaks are very difficult to remove by heat treatment.

6. The results of one of the ductility tests, which the Bureau of Steam Engineering specifies for the inspection of boiler tubes, are shown on Plate II. Two pieces of the metal in its original condition were flattened in a vice; the results are shown on the left of the plate. Three similar pieces were then heated in a furnace at 1,600 degrees F. for two hours, cooled in the furnace and flattened in the vice; the results are shown on the right of the plate. It will be noticed that the two pieces on the left both show cracks at the point of greatest bend, while the three heat-treated pieces show no cracks.

DISCUSSION.

The chemical analysis shows that the composition of this material may be considered satisfactory, although it would be better if the percentages of carbon and phosphorus were somewhat lower. The Bureau of Steam Engineering specifies no percentage limits for these constituents in boiler tubes. The American Society for Testing Materials, however, in specifications adopted June 1, 1912, gives 0.15 and 0.04, respectively, as the highest allowable percentages of carbon and phosphorus. These limits are reached or slightly exceeded in the tubes under discussion. Since both carbon and phosphorus decrease the ductility of steel, evidently the phosphorus content should be kept as low as possible and the carbon content should be kept as low as is consistent with the strength required.

As a result of the metallographic examination, it is evident that most of the specimens examined show distortion of the ferrite grains and more or less decided segregation of ferrite and pearlite. In some regions the pearlite occurs in scattered masses, elongated in the direction in which the tube was drawn; in other regions there are well-defined alternating streaks of ferrite and pearlite. The quantity of pearlite also varies considerably in different parts of the tubes.

The grain distortion and much of the streaked segregation are evidently due to cold working followed by insufficient annealing. Some of the segregation, however, must undoubtedly be traced back to segregation in the billets from which the tubes were made. Careful annealing will remove grain distortion and much of the streaked segregation; broad streaks cannot be removed by any practicable heat treatment.

The effects of cold working on the physical properties of steel are well known. The brittleness of these tubes, as shown by their behavior when the ends were expanded, is evidently due to the distorted and segregated structure caused by cold working. These conclusions, reached as a result of the metallographic examination, are confirmed by the results of the physical tests described in substance in the foregoing. According to the specifications of the Bureau of Steam Engineering, "Attest piece cut from each end of each tube must stand flattening under a press or hammer * * * without showing any defects on the outside." By reference to Plate II, it will be noticed that the two pieces on the left of the plate did not meet these requirements; the three heat-treated pieces, however, were flattened without showing any defects on the outside. It seems evident, therefore, that if these tubes had been more thoroughly annealed their brittleness would have been removed and the ends could have been expanded without cracking.

CONCLUSIONS.

The chemical analysis has shown that the composition of this material may be considered satisfactory, although it would be better if the percentages of carbon and phosphorus were somewhat lower.

Nearly all of the metallographic specimens have shown grain distortion and more or less decided segregation of ferrite and pearlite. This structure is caused by cold working followed by ineffective annealing. Since such treatment causes brittleness, the brittleness of these boiler tubes is therefore explained. The fact that samples from the tubes, when carefully annealed at the Experiment Station, lost most of their brittleness. lends additional support to this explanation.

HULL PLATE FROM DESTROYER.

A metallographic examination of three samples from a hull plate on one of the recent destroyers was made. The plate had failed on the trial trip of the destroyer. It cracked in a transverse direction and partly opened up. The percentages of sulphur and phosphorus in the metal are not given and no special heat treatment was given this metal.

The purpose of the test was to make a metallographic examination of the metal and determine its quality.

Plate I shows a photograph of the three steel samples. Samples marked 5, 6 and 12 were then cut into smaller pieces and polished on various surfaces for metallographic examination. Plate II is a photostat copy of a sketch showing the location of these metallographic specimens. Thirteen photomicrographs showing the typical structure of the material are shown on cards 1 to 13.

RESULTS.

In the photomicrographs of etched specimens, the white areas represent ferrite and the dark areas represent pearlite.

Card 1 shows the unetched surface of specimen 4, at a magnification of 45. Many slag particles are scattered over the surface. Some of the slag particles occur in irregular lines extending in the direction in which the plate was rolled.

Card 2 shows an edge of specimen 5 at a magnification of 100. A fine granular structure is found in this specimen. There are thin streaks of ferrite and pearlite extending in the direction in which the plate was rolled; these streaks, however, being very slight, would not injure the properties of the metal.

Card 3 shows the structure of specimen 7 at a magnification of 100. A fine-grained, uniform structure is found in this specimen.

Card 4 shows the unetched surface of specimen 9 at a magnification of 45. The area shown is near the surface of the break. A crack, extending from the surface of the break, appears in this photograph; a short crack nearly parallel to the longer one is also seen. These fine cracks follow lines of segregation of slag particles.

Cards 5 and 6 show adjacent areas on the unetched surface of specimen 12. The magnification in each case is 45. On card 6 the line of break is visible in the upper part of the photograph. Many small slag particles are scattered over the surface of the metal. Groups of blow holes, with cracks passing through them, also occur in this specimen; the cracks are parallel to the general direction of the line of break.

Card 7 shows a portion of the same view that is shown on card 6. The magnification is 100. Large blow holes, some of which are starting points for cracks, are plainly visible in this photograph.

Cards 8 and 9 show the etched surface of specimen 12 adjacent to the surface of the fracture; the area includes part of that shown on card 5. Card 9 shows the central part of the area shown on card 5. The

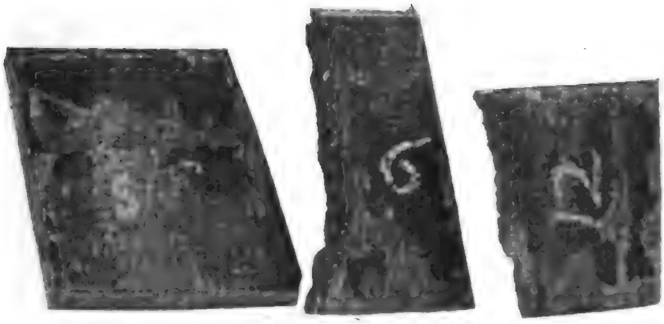
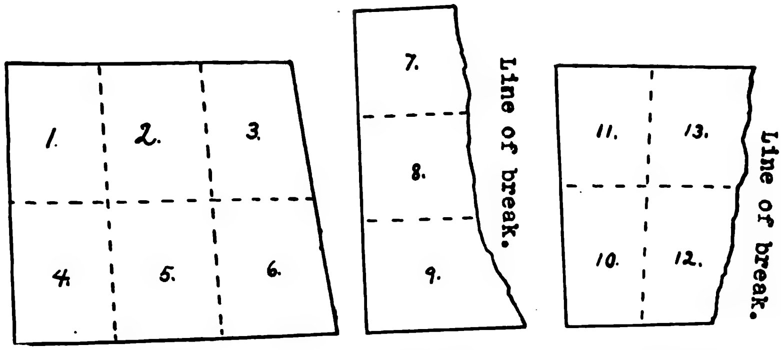


PLATE I.



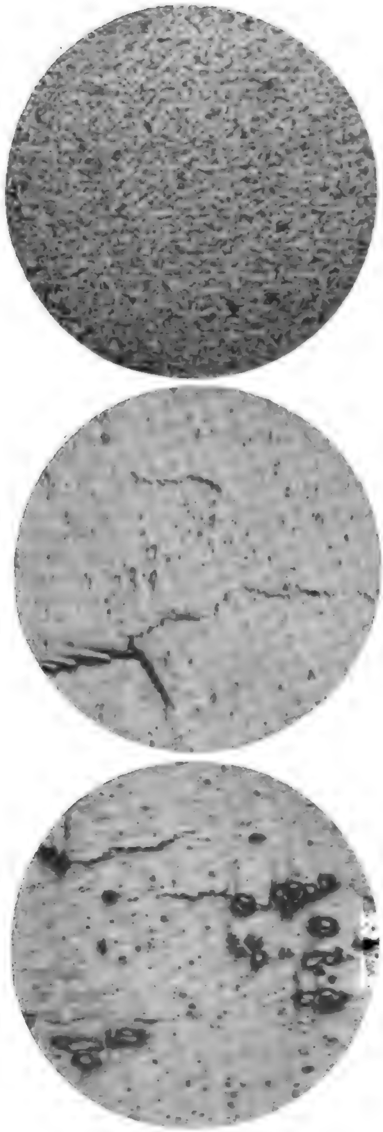
Sample 5.

Sample 6.

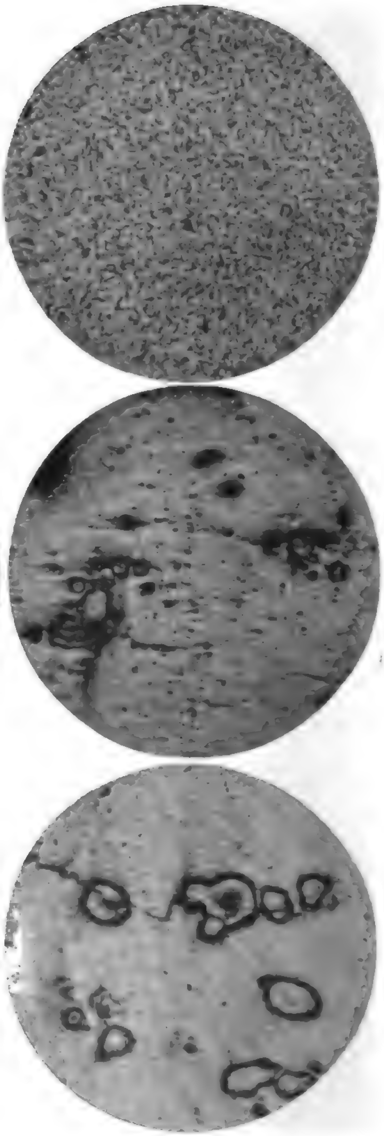
Sample 12.

PLATE II.

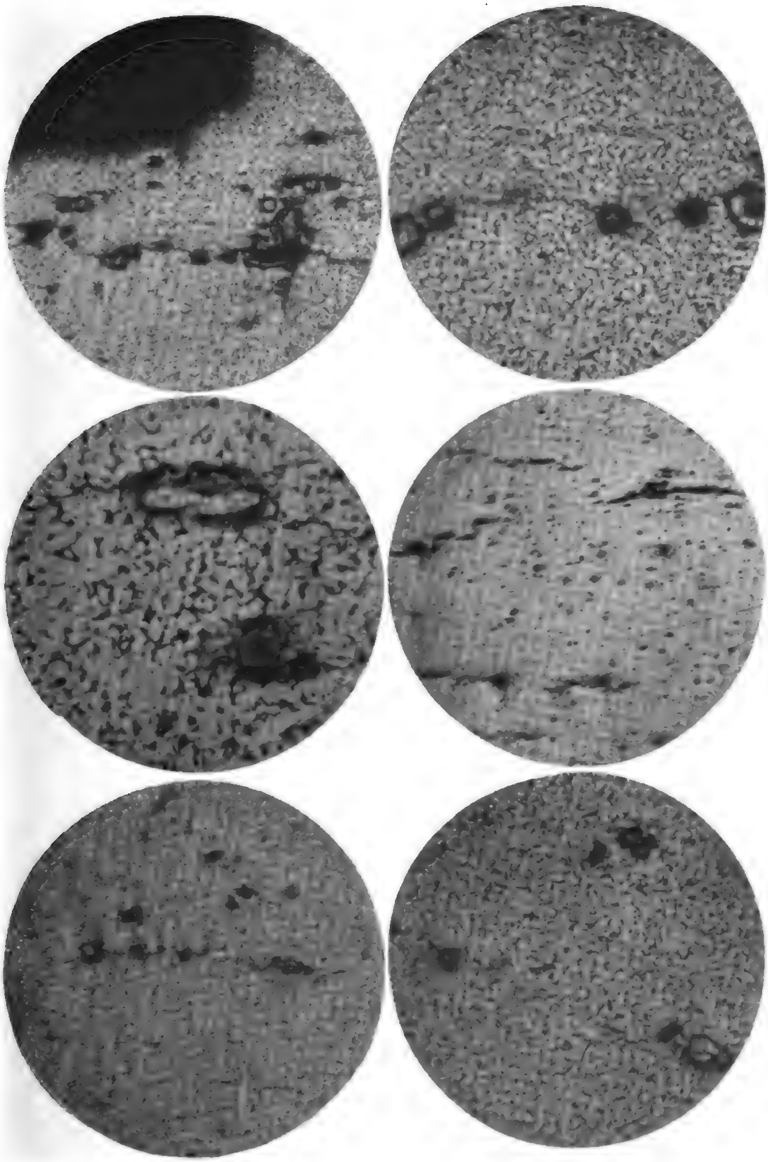
Card 1.—Unetched.



Card 2.
4.—Unetched.
6.—Unetched



Card 3.
5.—Unetched.
7.—Unetched.



Card 8.
10.
12.

Card 9.
11.—Unetched
13.

general structure of the metal is fine grained and uniform. Many blow holes, however, are visible on the surface; extending through these are incipient cracks parallel to the general direction of the surface of fracture.

Card 10 shows a portion of the etched surface of specimen 12 about ¼ inch from the surface of fracture. The magnification is 200. Two cavities in the metal are shown in this photomicrograph. Both of these cavities are probably blow holes, although one of them might be a hole left by the removal of slag during polishing. Minute cracks originate at both of the holes.

Card 11 shows the unetched surface of specimen 13 about ⅛ inch from the break. The magnification is 45. Many small slag particles and a few blow holes are scattered over the surface of the metal. Incipient cracks are visible starting from blow holes and slag particles and extending parallel to the surface of the fracture.

Cards 12 and 13 show the etched surface of specimen 13 about ½ inch from the surface of fracture. Card 13 shows, at a higher magnification, a portion of the same area that is seen on card 12. The etching has somewhat obscured the small slag particles, but groups of blow holes are visible. Some of the blow holes are starting points for cracks which are parallel to the surface of the fracture.

DISCUSSION.

The structure of the metal itself, as shown by the metallographic examination, is fine grained and fairly uniform. The edges of the plate show a slight degree of streaked segregation due to the rolling process; the amount of this segregation, however, is not sufficient to injure the quality of the metal.

Though the structure of the metal itself is good, non-metallic enclosures are present in such amounts that the material is decidedly weakened. As shown on cards 1, 4, 5, 6 and 11, many small slag particles are scattered throughout the metal. Many of these occur in lines or groups extended in the direction in which the metal was rolled. That these slag groups are regions of weakness is shown by the microscopic cracks which frequently follow the lines of segregation.

The chief source of weakness in this material, however, is the presence of groups of blow holes. These are especially numerous near the surface of break. Many incipient cracks, starting from these blow holes and often following an irregular line of blow holes, are found in this material. These cracks are invaribly parallel to each other and to the general direction of the surface of fracture.

CONCLUSIONS.

Though the general structure of the metal is good, the material is much weakened by the presence of non-metallic enclosures. Many fine particles of slag are scattered throughout the metal, and are often found in groups which are regions of weakness. The chief defect in the material, however, is the presence of groups of blow holes. These blow holes are especially numerous near the surface of fracture and were undoubtedly the chief cause of the failure of the plate.

U. S. S. *TACOMA*, PROPELLER SHAFT.

Examination was made of a section from the broken propeller shaft. The shaft had broken just abaft the after strut, at the end of the composition sleeve and the beginning of the taper. The section received for examination included the face of the fracture; this section including the composition sleeve is shown in Plates I and II. The appearance of the section after removal of the sleeve is shown in Plate III.

This shaft was manufactured by the Bethlehem Steel Company under specifications requiring high grade material. It was forged by press from a 35-inch round, fluid-compressed ingot; the shaft was annealed, tempered and then twice annealed. Details of heat treatment are not available.

The purpose of the test was to make a thorough examination of the material in order to determine its quality and, if possible, the cause of the fracture.

After the photographs (Plates I and II) were taken the composition sleeve was removed and the material was cut into sections as indicated by the dotted lines on Plate III. Samples for chemical analysis were taken and pieces were cut for physical tests and metallographic examination. A number of specimens were heat treated at the Experiment Station and subjected to physical tests. Sulphur prints were also made and a photograph of one of these is included in this report as Plate VI.

Results.—The results of the chemical analysis were as follows:

	Number of determinations.	*Average percentage.*
Carbon	2	0.348
Sulphur	2	0.0314
Manganese	2	0.724
Silicon	1	0.265
Phosphorus	1	0.021
Nickel	2	3.22

The results of tensile, bending, torsion, impact and endurance tests are given in Tables I to V. Both longitudinal and transverse test specimens were used. Some of these were heat treated at the Experiment Station. The heat treatment used was as follows:

Heat treatment (1).

Heated slowly to 1,700° F.; maintained for 1 hour; quenched in oil.
Reheated slowly to 1,425° F.; maintained for 1 hour; quenched in oil.
Reheated slowly to 1,125° F.; maintained for 1 hour; cooled in furnace.

Heat treatment (2).

Heated slowly to 1,700° F.; maintained for 1 hour; quenched in oil.
Reheated slowly to 1,425° F.; cooled in furnace.

The physical test specimens are designated by a letter showing the section from which the specimens were taken as seen on the photograph. Plate III.

The specimens used for metallographic examination were as follows:

Section A, 24 specimens.
Section B, 34 specimens.
Section C, 9 specimens.
Section D, 50 specimens.

Of these specimens, 13 photomicrographs were made showing the typical structure. In the six photomicrographs showing etched surfaces the white areas represent ferrite and the dark areas sorbite.

Cards 1 and 2 show two views of a longitudinal surface of specimen A17. The magnification in each case is 45. On the unetched surface are many slag particles arranged in lines extending lengthwise with the shaft. On card 1 a region near the fracture is shown; on the surface is a crack which extends away from the fracture.

Cards 3 and 4 show two views of a longitudinal surface of specimen A21 at a magnification of 45. Groups of many slag and sulphide particles are visible on the unetched surface. On card 3, which shows a region at the line of break, it will be noticed that the slag groups are especially numerous along the line of fracture.

Card 5 shows a longitudinal surface of specimen B30 at a magnification of 45. On the etched surface is a moderate sized network consisting of

PLATE I.

PLATE II.

PLATE III.

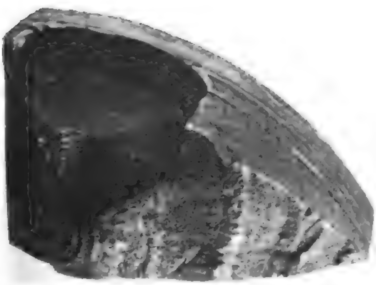


PLATE IV.

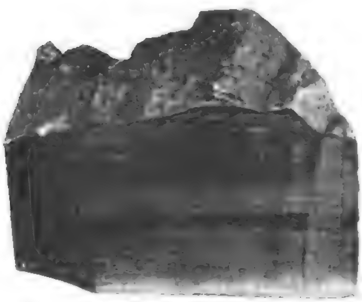
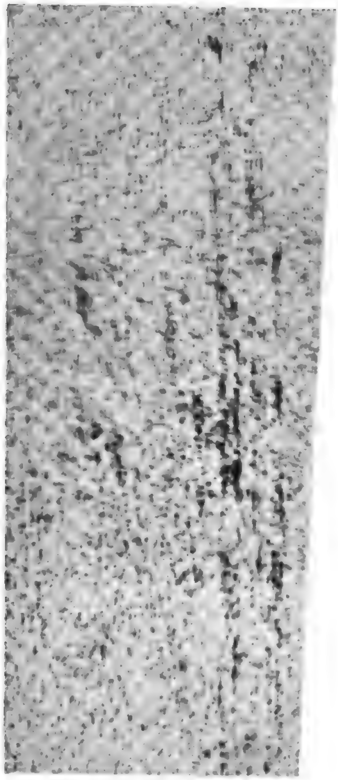
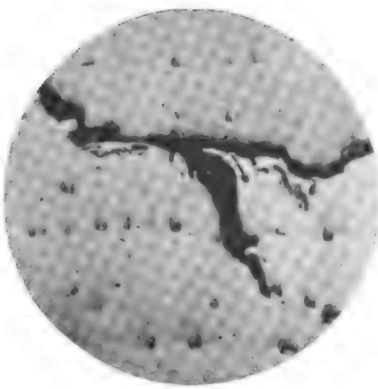


PLATE V.

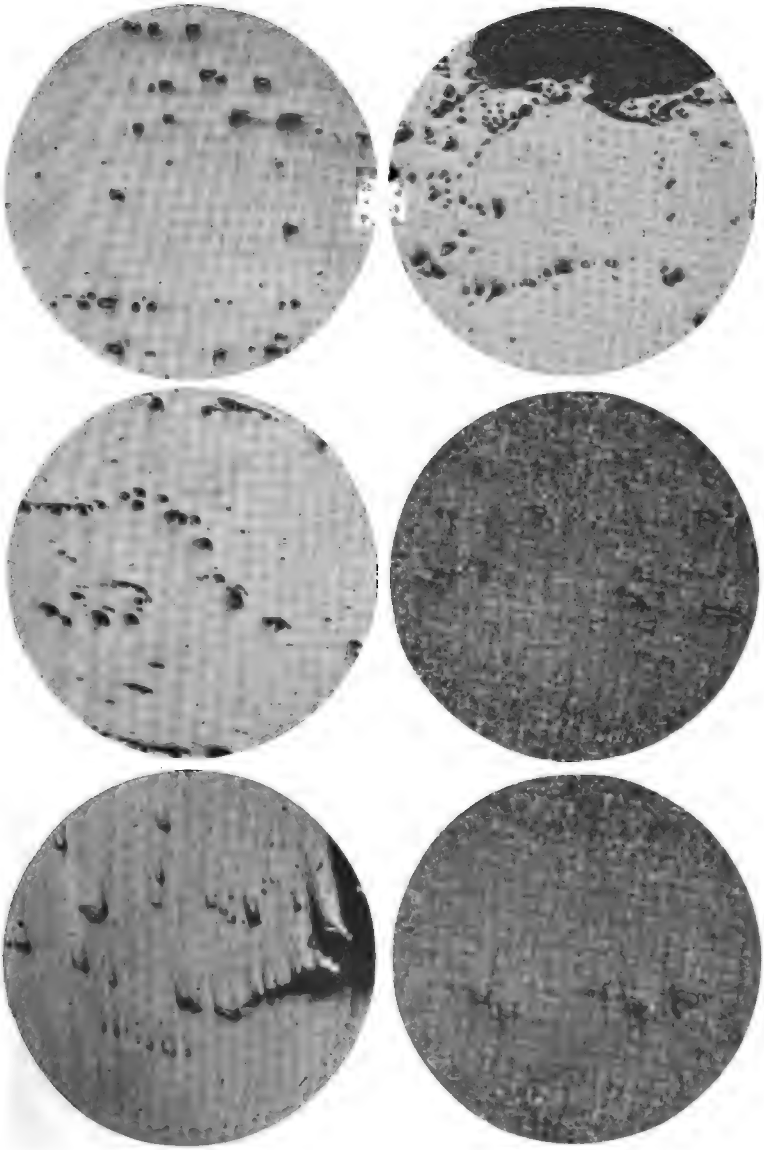


U. S. S. Tacoma.—Propeller Shaft.

PLATE VI.

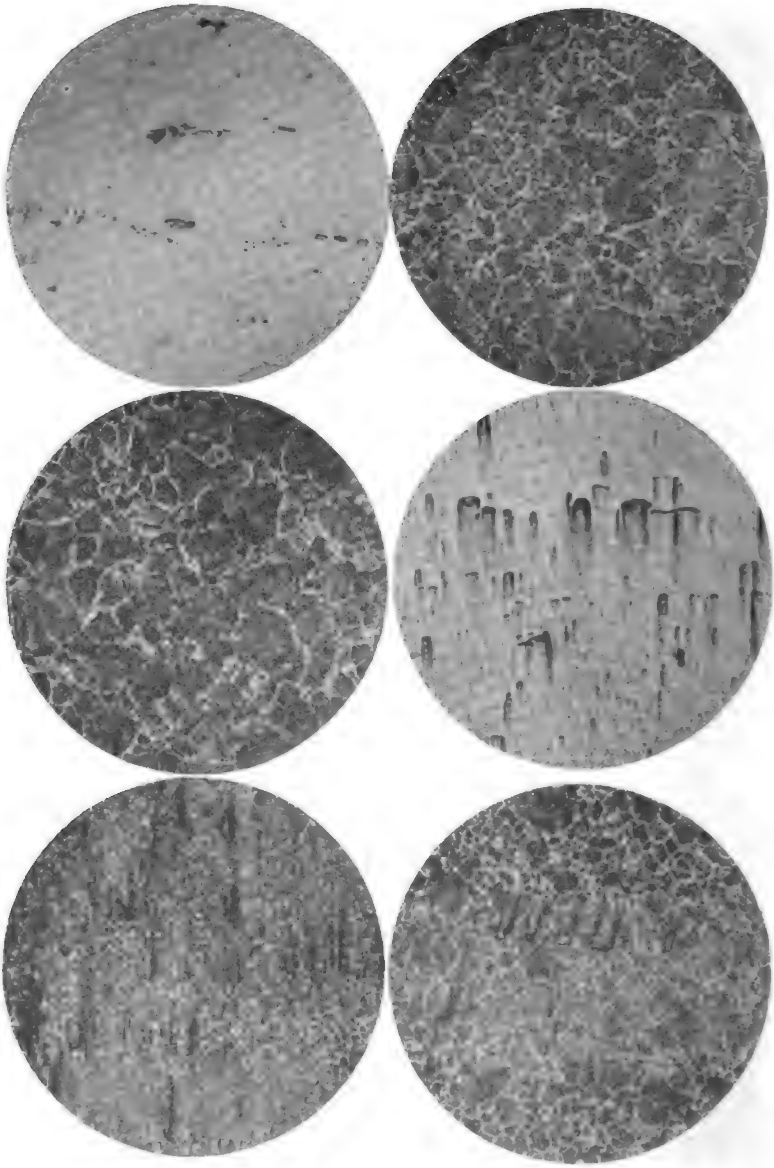


Card 1.—Unetched.



Card 2.—Unetched.
4.—Unetched.
6.—Unetched.

Card 3.—Unetched.
5.
7.



Card 8.—Unetched.
10.
12.

Card 9.
11.—Unetched.
13.

ferrite and sorbite. The view here shown is about two inches from the surface of break. In the middle of the photograph is a longitudinal streak lighter than the rest of the surface and containing many elongated particles of slag and manganese sulphide. Evidently the carbon content is below the average in this streak. Many such streaks were noticed in this material.

Cards 6 and 7 show two views of a longitudinal surface of specimen B34. The magnification in each case is 45. Card 6 shows the unetched surface at the line of break. Lines of slag particles, such as are seen on cards 1 to 4, are visible on the surface of this specimen. A short crack extends from the surface of break into one of these slag lines. On card 7, which shows the surface of the same specimen after etching, a broad streak having less than the average carbon content is shown. The structure is like that seen on card 5. In the streak are groups of slag particles.

Card 8 shows a longitudinal surface of specimen D35 at a magnification of 45. The view here shown is distant from the fracture. Lines of many small slag particles are seen on this surface.

Cards 9 and 10 show etched surfaces of specimens D38 and D46. The magnification in each case is 100. As on cards 5 and 7, a moderate-sized network is here shown. On card 9 a few elongated slag particles are visible.

Cards 11 and 12 show two views of a longitudinal surface of specimen D49. In both of these photomicrographs many segregated slag particles are visible. They show more clearly, however, on the unetched surface as seen on card 11. On card 12 it can be seen that there is segregation of ferrite and sorbite corresponding to the slag segregation.

Card 13 shows a longitudinal surface of specimen D50 at a magnification of 45. A longitudinal streak of less than average carbon content is seen in this specimen. This streak, with its lines of slag, is similar to those shown on cards 5 and 7 and to many other streaks found in this metal.

DISCUSSION.

As shown on cards 5, 7, 9, 12 and 13 the average grain size of this material is moderate. The metal, however, as shown on cards 1, 2, 3, 4, 6, 8 and 11, contains much slag. The slag particles are arranged in lines extending lengthwise with the shaft; they are especially numerous near the surface of the break, and, as shown on card 3, the fracture often follows these lines of segregation. That these slag streaks contain considerable manganese sulphide is indicated by Plate VI, which is a photograph of a sulphur print. Many lines of sulphid are visible on this print; their direction is lengthwise with the shaft.

Corresponding to the slag streaks are many broad bands having less than the average carbon content; a good example of such banks is shown on card 13.

Such segregation of slag and carbon is undoubtedly a source of weakness in the metal. It should not be assumed, however, that this alone caused the fracture of this shaft to take place. The quality of the metal is only one of a number of factors on which the endurance of the shaft depended. Plates 3, 4 and 5 show that the break originated at a hole in the shaft which was tapped out to receive a set screw. Ridges on the fractured surface radiate from the bottom of this screw hole, showing clearly that the break started from this point. Several examples of fractures, passing through such screw holes in shafts have been received at the Experiment Station. The weakening of the shaft by these holes is out of all proportion to the amount of metal removed; they should, therefore, be avoided whenever possible.

To determine the physical qualities of this metal a number of physical tests were made, the results being given in the foregoing. Tensile, bend-

ing, torsion and impact tests, made as usual with longitudinal specimens. are given in Tables I, III, IV and V. It will be seen that these longitudinal specimens gave very good results when subjected to all the tests except the single-impact tests given in Table V. The values obtained by the single-impact tests are less than they should be for a material of this composition; the metal, even after annealing, shows brittleness when subjected to sudden shock.

It is not customary to make physical tests with transverse specimens from shafts. Yet in shafts the transverse strength and ductility of the metal are of importance, though of less importance than in ordnance. The relation between the physical properties of longitudinal and transverse specimens varies considerably with the composition of the metal and the treatment it has received. It was thought, therefore, that valuable results might be obtained by making physical tests on transverse specimens from this shaft.

As shown in Table I, the transverse specimens had much less elongation and reduction of area than the longitudinal specimens, while the tensile strength and elastic limit are about the same. The impact test given in Table V also showed some superiority of the longitudinal over the transverse specimens. Part of the inferiority of the transverse specimens is undoubtedly due to the longitudinal streaks of slag and sulphide. Even with very uniform material, however, there is often a great inferiority shown by transverse specimens.

Two kinds of heat treatment were used, the results of which have been given in the foregoing. In both of these a preliminary heating at 1,700 degrees was given to produce uniformity. In heat treatment (1) the process known as "double annealing" was used, while in heat treatment (2) the material was simply annealed and cooled slowly. It will be noticed in Table II that neither kind of heat treatment produced improvement in the transverse tensile specimens. In the longitudinal specimens, however, the yield point was increased about 50 per cent., the tensile strength and ductility remaining about the same. As shown in Table V, heat treatment (1) tripled the numerical values obtained with longitudinal impact specimens and more than doubled the values obtained with transverse specimens; heat treatment (2), however, had very little effect on either longitudinal or transverse specimens. These results seem to show the superiority of "double annealing" over single annealing or "softening." It is not claimed, however, that this superiority would be found in an equal degree after the heat treatment of large objects.

More data as to the relation between the physical properties of longitudinal and transverse specimens, for different kinds of steel, will be obtained as opportunity arises. The relation between the physical properties produced by different kinds of heat treatment will also be the subject of investigation when possible.

CONCLUSIONS.

As shown by the metallographic examination, the grain size of this metal is moderate and the distribution of ferrite and sorbite is fairly uniform. There is, however, much slag and manganese sulphide segregated in groups and in lines extending lengthwise with the shaft. These slag particles are especially numerous near the surface of the break, and the fracture often follows the lines of segregation. Corresponding to these slag lines are many broad streaks having less than the average carbon content. This segregation of slag and carbon is undoubtedly a source of weakness in the metal.

In physical tests, longitudinal specimens, with the exception of impact specimens, gave good results. The impact specimens gave results which are too low for material of this composition; the metal, therefore, displays some brittleness toward sudden shock. Transverse tensile specimens

were deficient in ductility. Transverse impact specimens also showed much brittleness.

When heat treated at the Experiment Station longitudinal tensile and torsion specimens showed much increase in elastic limit alone, while transverse tensile specimens showed little or no improvement of any kind by heat treatment. Impact specimens showed much improvement by "double annealing" but practically no improvement by single annealing or "softening." It seems evident, therefore, that the brittleness toward sudden shock is not due to "hardness" of the metal.

Though the metal has much slag segregation, and displays brittleness, particularly in a transverse direction, it should not be assumed that the break was entirely due to the defects of the metal. The screw hole shown in Plates 2, 3 and 4 undoubtedly weakened the shaft somewhat. It is impossible, therefore, to say whether or not the shaft would have broken if the metal had been better and other conditions had remained the same. It is equally impossible to say whether or not the shaft would have broken if the screw hole had not been there and other conditions had remained the same.

TABLE I.

TENSILE TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

	Longitudinal.			Transverse.			
Standard Specimen No.	E2	E7	Ave.	B2	D2	D4	Ave.
Diameter:							
Test piece, in.	.505	.505		.505	.505	.505	
Fracture, in.	.352	.355		.458	.468	.432	
Area:							
Test piece, sq. in.	.200	.200		.200	.200	.200	
Fracture, sq. in.	.097	.099		.164	.168	.146	
Reduction, pr. c.	51.5	50.5	51.0	18.0	16.0	27.0	20.3
Length between punch marks in inches	2.	2.		2.	2.	2.	
Elongation:							
Inch	.47	.47		.17	.31	.26	
Per cent.	23.5	23.5	23.5	8.5	15.5	18.0	14.0
Yield Point:							
Pounds	11,300	13,100		12,150	12,700	12,800	
Pounds per sq. in.	56,500	65,500	61,000	60,750	68,500	64,000	62,750
Maximum Stress:							
Pounds	21,080	21,060		20,100	21,200	21,280	
Pounds per sq. in.	105,400	105,300	105,350	100,500	106,000	106,400	104,300
Remarks:							
Fracture	medium	medium		coarse striated	coarse striated	coarse striated	

TABLE II.

TENSILE TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

Heat-Treated Specimens.

	Longitudinal.			Transverse.		
Standard Specimen No.	E11	B1	B3	D10	D8	Ave.
Diameter:						
Test piece, in.	.505	.505	.505	.505	.505	
Fracture, inch	.325	.460	.453	.430	.450	
Area:						
Test piece, sq. in.	.200	.200	.200	.200	.200	
Fracture, sq. in.	.083	.166	.161	.145	.159	
Reduction, per cent.	58.5	17.0	19.5	27.5	20.5	21:1
Length between punch marks in inches	2.	2.	2.	2.	2.	
Elongation:						
Inch	.46	.28	.20	.35	.24	
Per cent	23.0	14.0	10.0	17.5	12.0	13.4
Yield Point:						
Pounds	18,800	13,000	12,150	13,100	12,800	
Pounds per sq. in.	94,000	65,000	60,750	65,500	64,000	61,310
Maximum Stress:						
Pounds	23,250	21,000	18,950	21,100	21,100	
Pounds per sq. in.	116,250	105,000	94,750	105,500	105,500	102,690
Heat treatment	1	1	2	1	1	
Remarks:						
Fracture	fine	coarse striated.		coarse striated.		

TABLE III.

TORSION TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

	Longitudinal Specimens.	
Standard Specimen No.	E1	E13*
Diameter of piece, inch	.750	.750
Length between punch marks, inches	3.	3.
Angle of twist at elastic limit, degrees	1.63	2.2
Angle of twist at maximum torque, degrees	580	660
Elastic limit, pounds inches	3,200	4,400
Maximum torque, pounds inches	8,150	9,030
Shearing stress at elastic limit, pounds per sq. in.	38,600	53,100
Nominal shearing stress at maximum torque, pounds per sq. in.	98,400	109,000
Modulus of transverse elasticity, pounds per sq. in.	10,800,000	11,606,000

*Heat treated at the Experiment Station.

TABLE IV.

BENDING, ALTERNATING-IMPACT AND ENDURANCE TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

		Longitudinal,	
Bending Test:			
Standard specimen			E5
Degrees of bending without breaking			180
Alternating-impact test:			
Standard specimen	E4	E10*	E13*
Fall of hammer, centimeters	9	9	9
Number of falls required to break	507	632	643
Endurance test:			
Standard specimen	E3		E6
Load used, pounds	135		135
Fiber stress (353.38 times load) pounds	47,706		47,706
Ratio, fiber stress to elastic limit, per cent	78.2		78.2
No. of revolutions required to break	619,800		487,000

*Heat treated at the Experiment Station; Heat treatment (1).

TABLE V.

SINGLE-IMPACT TESTS.—PROPELLER-SHAFT FORGING, U. S. S. "TACOMA."

Direction.	Standard specimen ¼″ x ⅜″ cross section.	Energy required to break Ft. pounds.	Heat treatment.
Longitudinal	B4	24	Untreated
"	B5	27	"
"	B6	77	(1)
"	B7	76	(1)
"	C3	74	(1)
"	C4	78	(1)
"	C5	77	(1)
"	C6	81	(1)
"	C7	28	(2)
"	C8	31	(2)
"	C1	41	(2)
"	C2	42	(2)
Transverse	D8	16½	Untreated
"	D9	21⅛	"
"	C11	54	(1)
"	C12	52	(1)
"	C13	54	(1)
"	C14	47	(1)
"	C9	20½	(2)
"	C10	14	(2)

SUBMARINE ENGINES.

By A. P. Chalkley, B.Sc., A. M. Inst. C. E.

Whilst it is true that the development of the submarine in its present form would have been almost an impossibility had it not been for the internal-combustion engine, it is even more apparent that the capacity of the later type to play an important rôle in offensive warfare is due almost entirely to the great progress that has been made during recent years in the construction of the Diesel oil engine. It is true that in certain cases (for instance, in some of the French submersibles) steam turbines are employed for propulsion on the surface, but these exceptions are not numerous and general opinion is unfavorable towards such craft. At any rate, they do not appear to have been so successful in practice as the boats in which the surface propelling machinery consists of oil engines.

All the earlier submarines built in this country and in Germany, as well as most of those in France, were equipped with petrol or paraffin engines, but it was very soon apparent that the employment of such motors put a very distinct limitation upon the possibilities of the submarine. Quite apart from the danger involved, which was by no means negligible, the construction of petrol motors of high power was not to be considered for this work, and above all, owing to the relatively high fuel consumption, the radius of action of a petrol-engined submarine was too small to enable the boat to take part in extensive operations, even were it suitable in other ways. Yet, full credit should be given to the petrol-engine manufacturers who ten years ago undertook to construct engines of 200 or 300 H.P. and built them so well that, even now, several of them are still doing very good work. In most of the early British craft a horizontal opposed type was employed, but in many of the foreign submarines vertical motors were installed, and Fig. 1 shows such a machine built by Messrs. Thornycroft for the Italian Navy about 1905 or 1906.

The first Diesel-engined submarines were completed seven or eight years ago, and there was very little difference between the times when the earliest British, German and French craft with the new propelling machinery were commissioned. Probably the idea originated in France, but the earlier boats were about four years under construction, so that it was, if anything, later before they were put into service. Far more difficulties were encountered in the manufacture of these motors than was anticipated, the troubles being mainly due to the high speed of rotation which was necessary in order to reduce the weight of the machinery to a minimum and to diminish the size in view of the limited space available for the installation. The construction of the air compressors for the injection of the fuel was in itself no mean problem, and it was also very difficult for the designers to overcome the troubles which arose in consequence of the high temperature of combustion. This may be in the neighborhood of 1,200 to 1,500 degrees centigrade, and as the number of explosions per unit of time is much greater than in a slow-running engine, the heat problem is rendered more difficult of solution. It is not surprising, therefore, that in the earlier submarine engines, there were many accidents owing to inefficient piston cooling, and the employment of steel pistons (with the object of gaining greater strength) working in cast-iron cylinders—a method instituted by one or two French builders—did not improve matters and was quickly abandoned.

It is only necessary to quote the respective weight of slow-running Diesel engines as employed on mercantile vessels, and the high-speed sets used on submarines to show the vast difference there must be in the construction of the two types. In the first case the weight, including accessories, is generally in the neighborhood of 350 pounds per B.H.P., whilst submarine engines on an average do not weigh more than 50 to

70 pounds per B.H.P., usually the lower figure. It need hardly be pointed out that the greatest care has to be exercised in weight cutting in Diesel engines, as the excessive pressures that have to be withstood do not allow of any weakening of the main structural parts. It is for this reason that most of the German submarine engines are built largely of manganese bronze, which, although adding somewhat to the expense, is thought by the German designers to be the most satisfactory method.

The requirements of submarine engines which have just been mentioned—smallness and light weight—would naturally lead to the conclusion that the two-cycle motor should offer great advantages for the work. This is to a certain extent the case for a Diesel engine operating on the two-cycle principle should, apart from exceptional conditions, occupy less space and have a lower weight than the four-cycle engine developing the same power. In practice, however, the difference does not appear to be very large, and many engineers are of opinion that the greater difficulties involved in the construction of the two-cycle type do not warrant its adoption for the relatively small advantages gained.

It is a matter of particular interest that in Germany, although the four-cycle Augsburg motor was originally employed, the two-cycle Diesel engine is now invariably adopted for all submarines, this policy having been followed for some years past. In England, on the other hand, nearly all the machinery for the Diesel-engined submarines has been built by Vickers, who have employed the four-cycle principle, although their motors are distinctive in that they avoid the use of the air compressor for injection, since solid or mechanical injection of fuel is utilized. This naturally effects a saving in weight and space, and although the Vickers engines of 850 H.P. in the modern British submarines are eight-cylinder four-cycle machines, and some of the German submarine motors of about the same power have only six working cylinders and operate on the two-cycle principle, there is probably very little difference either in the weight or in the overall dimensions of the engine. This is not difficult to explain when it is considered that there are no injection-air pumps on the Vickers motors, whereas in the Krupp engines referred to there are two scavenger pumps and two air compressors. For some of the new British submarines, however, two-cycle engines have been constructed.

In France both two and four-cycle engines have been installed, some of each type having been built in Germany for the purpose, besides those constructed in France, but it is somewhat curious that in Sweden, where the Polar Co., of Stockholm, have in the past year or two taken up the manufacture of submarine engines, the four-cycle type has been adopted in spite of the fact that the success of this firm has been built up on the manufacture of motors of the two-cycle principle. In Switzerland Messrs. Sulzer build two-cycle submarine engines, whilst in Italy a similar type manufactured by the Fiat Co. has been adopted in recently constructed submarines.

It would thus appear that, on the whole, the two-cycle engine has found most general acceptance and when the submarine develops into a much larger and faster craft than it is at present, it appears probable that the two-cycle type will have to be employed, owing to its general suitability for higher powers. Owing to the complications involved in making them directly reversible, most four-cycle submarine engines have hitherto been constructed as non-reversible sets, the electric motors used for underwater propulsion being employed for reversing. Exception to this should be made in the case of the Polar engine, which will be referred to later, but it may be mentioned, on the contrary, that practically all the two-cycle engines are directly reversible, and it can easily be understood that the design of a two-cycle reversible motor is much simpler than that of the four-cycle type unless some special means be adopted.

The differences in design of the various types of Diesel engines em-

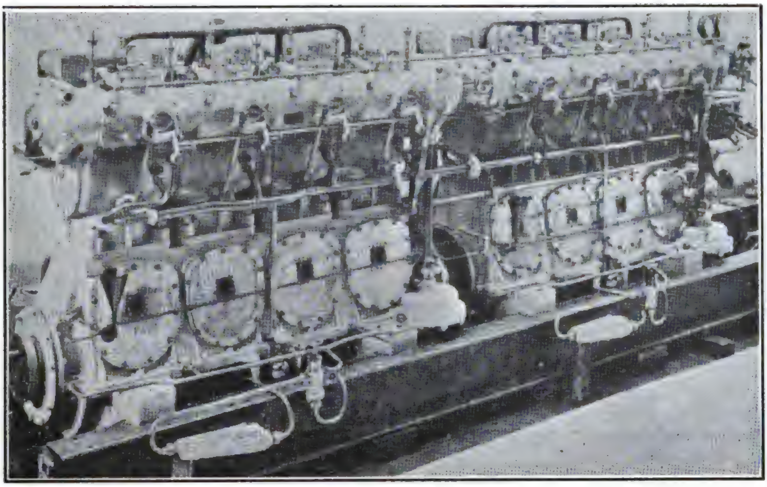


FIG. 1.—A THORNYCROFT SUBMARINE ENGINE BUILT EIGHT YEARS AGO FOR THE ITALIAN NAVY.

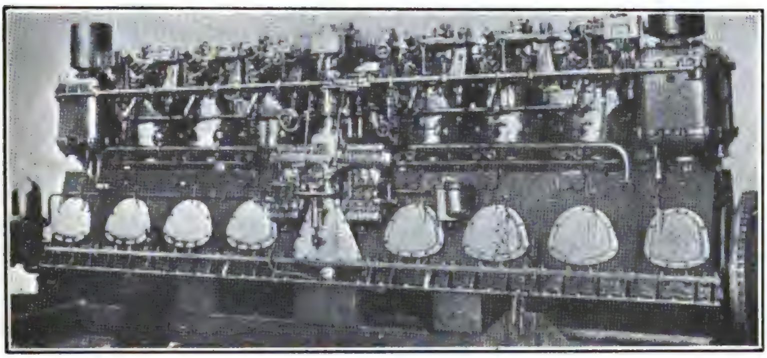


FIG. 2.—KRUPP SUBMARINE DIESEL ENGINE OF 900 B.H.P. FOR THE GERMAN NAVY.

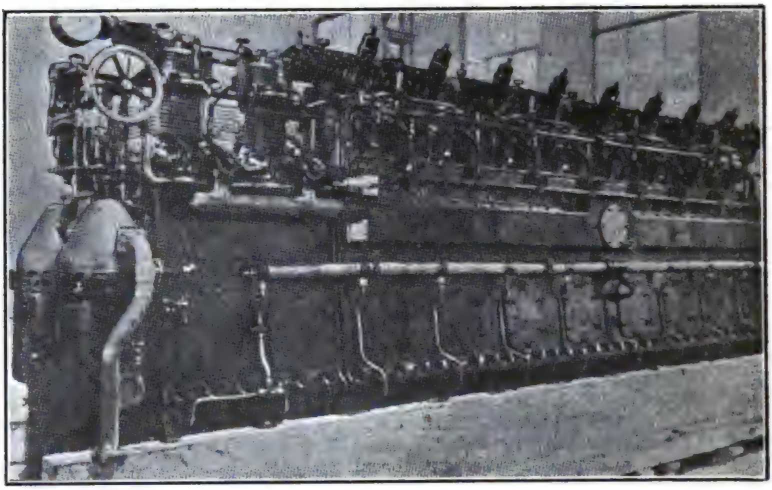


Fig. 3.—A 900 B.H.P. Nurnberg Submarine Motor.

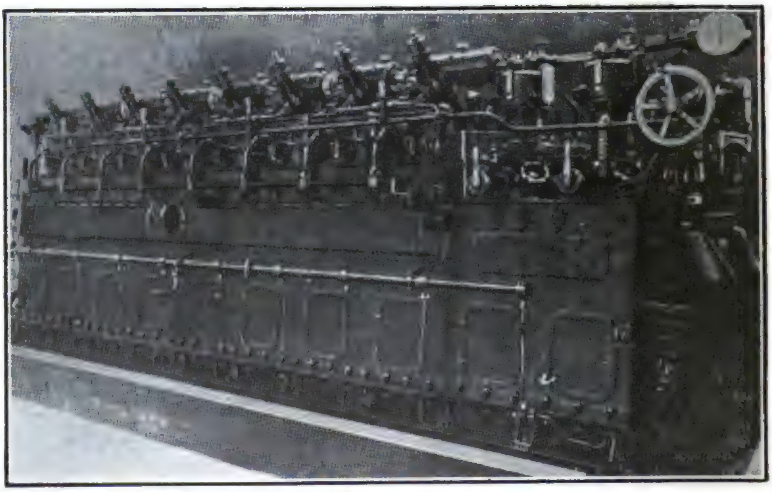


Fig. 4.—A Two-Cycle Nurnberg Submarine Motor.

ployed on submarines as built by the different manufacturers are quite remarkable, and the fact that practically all of them have done excellent work is a clear indication of the diverse manners in which a problem, with one end in view, can be worked out successfully. Nothing further need be said regarding the motors used on British submarines, but some details may be given of the types employed in the underwater craft of other Navies. In Germany, Krupps, of Kiel, and the Maschinenfabrik Augsburg-Nürnberg, build all the motors used in German submarines, and the former type is generally considered to be the more successful, although reports are somewhat at variance.

Fig. 2 shows one of the standard Krupp engine which is installed in many of the very latest German submarines, from U17 onwards, being designed to develop about 900 B.H.P. at 450 r.p.m. It is a two-cycle six-cylinder machine with two air compressors driven off the crankshaft, between the working cylinders, and two scavenger pumps also directly coupled to the crankshaft and arranged at the extreme ends of the engine. The great difficulty in the design of these high-speed two-cycle engines is to ensure an effective supply of scavenging air to the cylinder, and to cope with this trouble there are three scavenging valves in each cylinder cover operated from one cam on the camshaft by means of a linkwork. As these valves occupy practically the whole available space in the cover, the fuel-inlet and starting valves have to be set diagonally and the arrangement can be seen from the illustration. The engine is directly reversible and is, of course, practically enclosed, as are all the high-speed submarine Diesel motors.

The Nürnberg engine is in some ways more interesting than the Krupp machine. It is also designed for an output of about 900 B.H.P. at 450 r.p.m. and operates on the two-cycle principle. It has eight cylinders, each 310 mm. in diameter and 340 mm. stroke, and the two two-stage air compressors are driven directly off the end of the crankshaft. Figs. 3 and 4 show two opposite engines complete for installation in a submarine, and it will be noticed that the wheels and levers for controlling the engines are opposite each other when the motors are actually fitted in the vessel.

The chief point in the design of this engine (which is well illustrated in Figs. 5 and 6, the former being a sectional front elevation, and the latter the end elevations) is the fact that the scavenging pumps are arranged as stepped pistons immediately below the working cylinders, so that there are in all eight such pumps. It might be thought that this would give an excessive supply of air, but it must be remembered that the actual scavenging volume in each pump cylinder is not large, corresponding only to the difference between the diameters of the working and scavenge cylinders. The actual diameter is 475 mm., and the stroke of the pump is, of course, the same as that of the working piston, viz: 340 mm. The high-pressure cylinder of the compressors has a diameter of 100 mm., and the low-pressure 300 mm., the stroke for both being 250 mm. The overall length of this engine is under 23 feet, and the height is about 8 feet 6 inches.

The piston-cooling arrangement in the motor is somewhat unusual, and is carried out by pumping oil at a pressure of about 35 pounds per square inch through the crankshaft, and connecting rod, and thence to the gudgeon pin, after which it passes through a pipe to the body of the piston. It is then returned to the oil cooler and used over again; this method is interesting in contrast to the one employed on the previously-described motor, where water cooling is adopted. There is only one scavenging valve to each cylinder, and this is inclined to the axis of the cylinder, as is also the fuel-inlet valve on the opposite side. The method of reversing adopted in this case is to move the cam shaft through an angle of 30 degrees, which sets the scavenge valve and fuel valve at

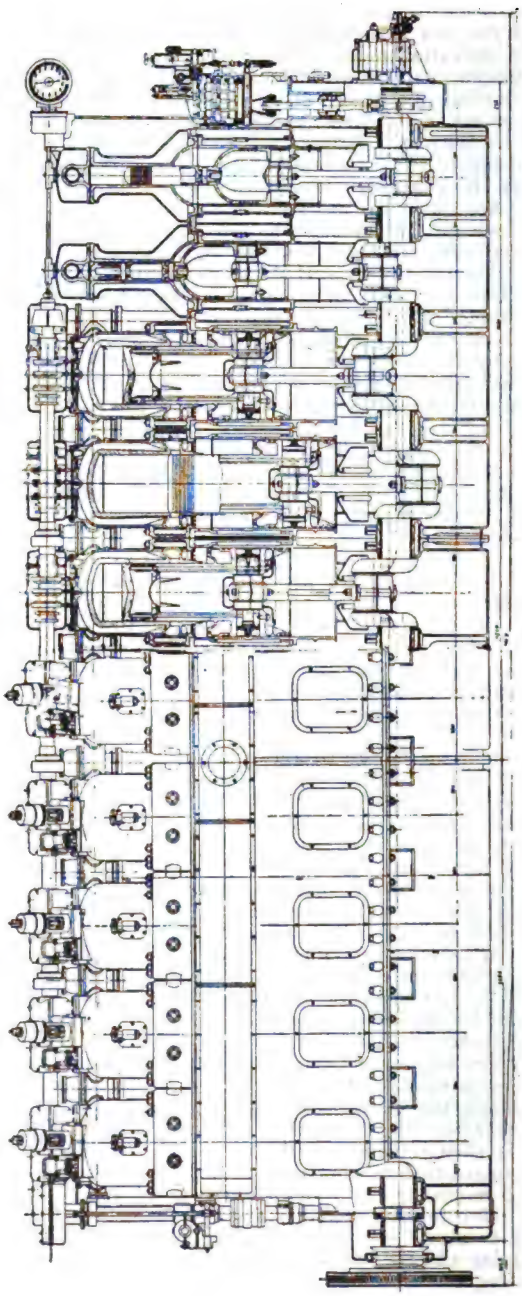


Fig. 5.—Front Elevation and Section of the 900 B.H.P. Nurnberg Engine Shown in Fig. 4.

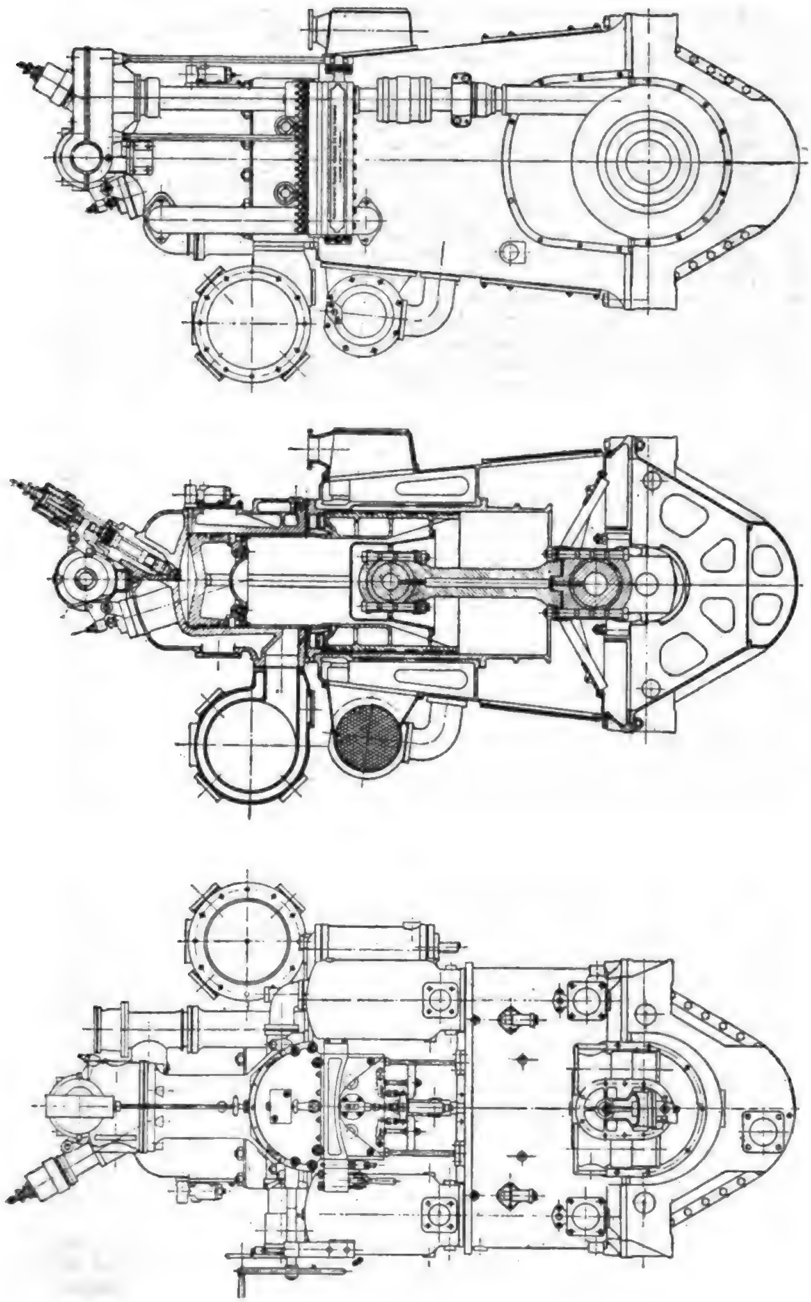


FIG. 6.—END ELEVATIONS AND SECTION OF A PAIR OF NURNBERG TWO-CYCLE DIESEL ENGINES.

the correct positions for running in the opposite direction. The design of the Fiat motor, built in Italy, is similar to this type, in that the stepped piston arrangement is also employed for the provision of scavenging air.

The Sulzer submarine engine is perhaps the most novel of any type constructed. It has been built for the American and Japanese navies as a six-cylinder machine, developing 600 B.H.P. at 400 r.p.m. It is also of the two-cycle single-acting type, but the main point of differentiation between it and the motors previously described is that instead of scavenging being carried out by means of valves in the cylinder head, it is provided for by means of ports at the bottom of the working cylinders. These can be seen in the sectional illustration of the engine at Fig. 9. The advantage derived by this arrangement lies in the fact that the cylinder cover has a smaller number of holes for the valves, and is therefore less liable to develop cracks, this being a frequent source of trouble with high-speed Diesel engines. It is only necessary for the fuel valve and the starting-air valve to be fitted, and, as can be seen from the illustration, there is ample space for these valves whilst allowing a very generous design of cylinder cover with good cooling spaces. Beyond this the operation of reversing is somewhat simplified.

It will be noticed that there are two sets of ports for the admission of scavenging air, the upper one being controlled by means of a mechanically-operated valve, actuated from the crankshaft by means of levers. The object of this design is for an extra supply of scavenging air to be admitted to the working cylinder after the main supply to the bottom port has cleared out most of the exhaust gases.

There are various other points of interest in this motor. It will be noticed that there is a firing plate, dove-tailed into the top of the piston, in order to take the effect of the high temperature of the flame—a method that has also been adopted by other manufacturers. The arrangement for supporting the cylinders is somewhat unusual for a submarine type of motor, steel columns being provided which reach right through the bedplate. There are also diagonal supports to ensure stiffness, and with this arrangement it is possible to have a very large crank-chamber door, which can be readily removed, and enable the piston to be taken out, if necessary, in a more convenient manner than is usually the case.

It has been thought that with this type of scavenging there would be some difficulty in constructing motors of high power, but this would not seem to be the case, since engines of 900 H.P., running at about 350 to 400 r.p.m., have already been built, whilst others up to 2,400 B.H.P. are in contemplation, if they have not already been put into construction. These motors, like others of the two-cycle type, are directly reversible, and an illustration of the complete motor ready for installation is shown in Fig. 7. The cylinders have a bore and stroke of 320 mm.

Mention was made earlier in this article to the four-cycle Polar submarine motor, and this is illustrated in Fig. 8, the machine shown being one of 350 B.H.P., running at about 500 r.p.m. It has six cylinders with a bore of 290 mm., and a stroke of 300 mm., and a two-stage air compressor is mounted on the same bedplate, being driven directly off the end of the crankshaft. It will be seen that externally the engine differs from the ordinary design, in that the camshaft is set lower than usual, necessitating long valve levers. The chief novelty in the construction lies in the arrangement adopted for rendering the motor directly reversible. In order to simplify this to the greatest degree it has been designed so that, in reversing, the exhaust valve and the inlet valve are changed over, the exhaust becoming the inlet and the inlet the exhaust.

During reversal the camshaft is moved longitudinally, in order that the reverse cams for the fuel-inlet valve may come into operation, but the whole method is certainly extremely simple for a four-cycle engine.

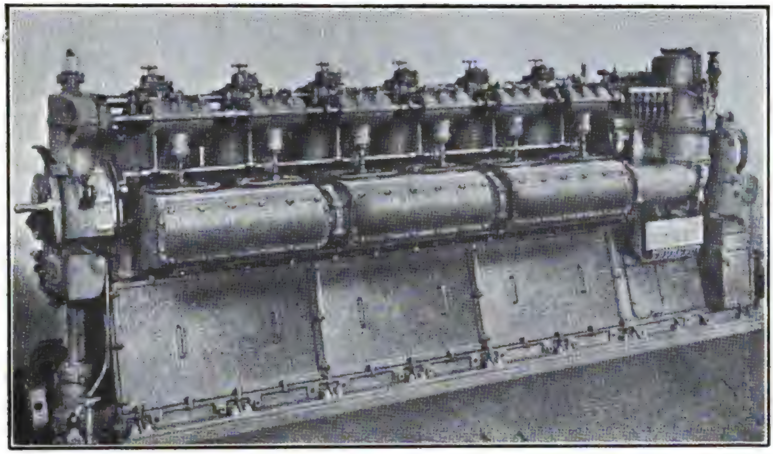


FIG. 7.—A SULZER TWO-CYCLE ENGINE OF THE TYPE USED IN THE AMERICAN AND JAPANESE NAVIES.

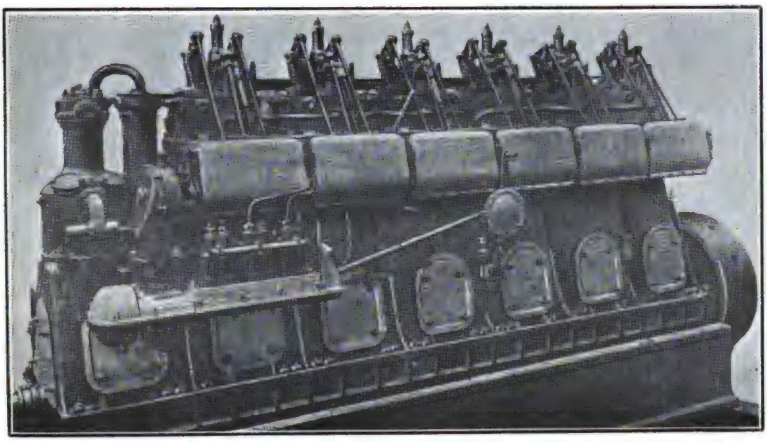


FIG. 8.—A POLAR FOUR-CYCLE ENGINE INSTALLED IN SWEDISH SUBMARINES.

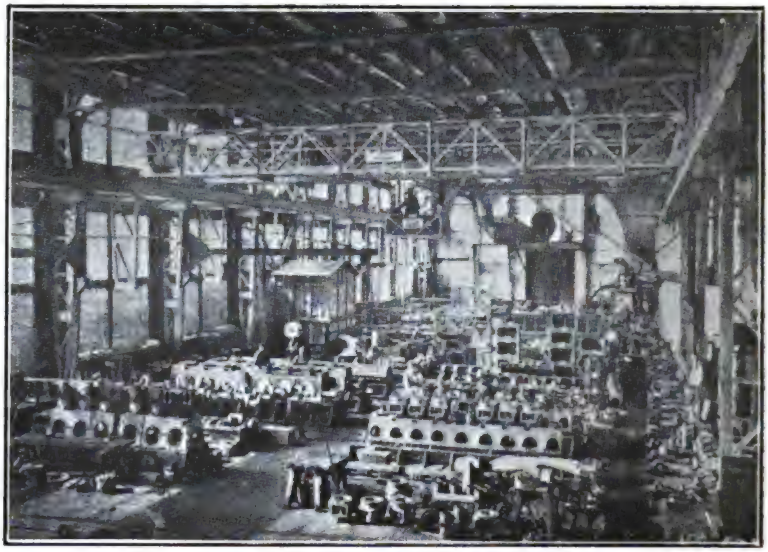


Fig. 10.—Krupp's Submarine-Engine Erecting Shop.

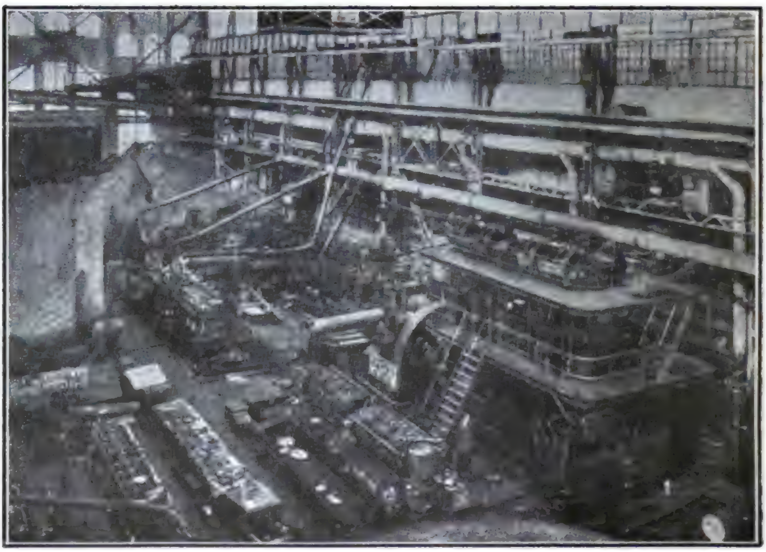


Fig. 11.—Submarine Engines Being Tested in the Shops at Krupp's Works.

Motors of this type are now built in fairly large sizes, at any rate, up to about 600 B.H.P.

It is no secret that, even before the war, all the leading naval powers were endeavoring to produce a submarine of much larger size, and with a higher speed than the types now being used, and the present war has, of course, only caused the naval authorities to hasten towards this end. The development of the submarine is only limited by the limitations of the oil engine, and immediately satisfactory motors of 2,000 or 3,000 B.H.P. can be produced, submarines of much larger size and with speeds of 20 knots or over will be constructed. The difficulties in the manufacture of such high-powered motors running at speeds of 400 r.p.m. are enormous, and a considerable period may elapse before complete success is attained.

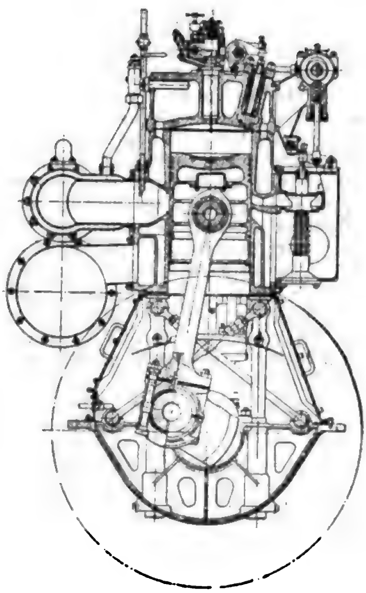


Fig. 9.—Sectional End Elevation of a Sulzer Two-Cycle Engine.

It must not be thought, however, that it is only the large type of submarines that is required in the future, for it is evident that the purely defensive submarine will figure very largely in naval programs. This is particularly the case in the smaller countries whose navies cannot hope to cope with those of the larger powers, but who may be able to maintain a very strong defensive by the employment of numerous small submarines. It is for this reason that a very great development of the submarine-engine industry may be expected in the next few years, for although the submarine may not, as has been suggested in some quarters, render the battleship obsolete, yet it is bound to play a very important part in all naval questions in the next 10 or 15 years.—"Cassier's Engineering Monthly."

GERMAN SUBMARINES.

The Italian naval review "Rivista Marittima" gives in its issue for March an abstract of an article written by a German officer, according to which the German submarine *U 47* completed her trials in February last. The number of submarines in the service at the opening of hostilities was 27. Adding those then in course of construction for Germany, the number reached 36. New units have been put down since the war was declared, besides those then in course of construction for foreign governments. The crews for submarines, when not in action, are quartered in hulks which are anchored in German waters. When a submarine leaves for a cruise she is so filled with stores and fresh water that only a very narrow passage remains free. Naphtha exudes at the steel bulkheads, doors, and throughout the whole ship, this lending confirmation to the rumor that the water-ballast tanks, which, though watertight, are not oiltight, are also filled with oil. The crew of the *U 47* and similar units number 30 men. The men's quarters are forward in the torpedo-launching room, and aft in the electric-motor room; the height of these rooms at the center line of the ship is about 3 m. (9 feet 10 inches). The same review deals elsewhere with the radius of action of the German submarines, and states the following: If we consider an 800-ton submarine. propelled by Diesel engines, and having a cruising speed of 8 knots on the surface, the power developed does not exceed 250 horsepower, and the oil-fuel consumption is at most 10 kg. per mile. On this basis, for every thousand miles the oil-fuel consumption is 10 tons. Considering that the boat probably carries a normal supply of 50 tons, she would have a radius of action of at least 5,000 miles, equivalent to over 600 hours of constant navigation at 8 knots. If, moreover, the water-ballast tanks be filled with oil, the very great increase in radius of action thereby made possible is obvious. In fact, the radius of action appears now to be limited only by the power of resistance of the crew.

SMALL CONDENSING TURBINES.

By W. J. A. London, Chief Engineer, the Terry Steam Turbine Co.

Since the introduction of the small direct-connected turbine on a commercial basis, some eight years ago, until recently, fully 90 per cent. of the machines called for were intended for noncondensing service. In the few cases where they were called upon to operate condensing, such as for marine work, little attempt was made at economy, as the operation of these machines condensing was more a matter of convenience than of water rate. It has been acknowledged that the designing of small turbines is much different from that of large machines, for were a small turbine designed on the same principles and lines as a big machine a hopeless commercial failure would result. There have, therefore, been two distinct fields in turbine work; the principles governing the designs of small and large machines being so much at variance that they might be said to be almost as different as the designs of a steam and a gas engine.

Most small turbines were installed to operate noncondensing, being used primarily for auxiliary apparatus, the exhaust being used in feed-water heaters. Small isolated plants were operated noncondensing, the exhaust steam being used for industrial or heating purposes.

For isolated plants, such as small pumping installations of, say 150 to 500 H.P., where economy was of much importance and the saving by

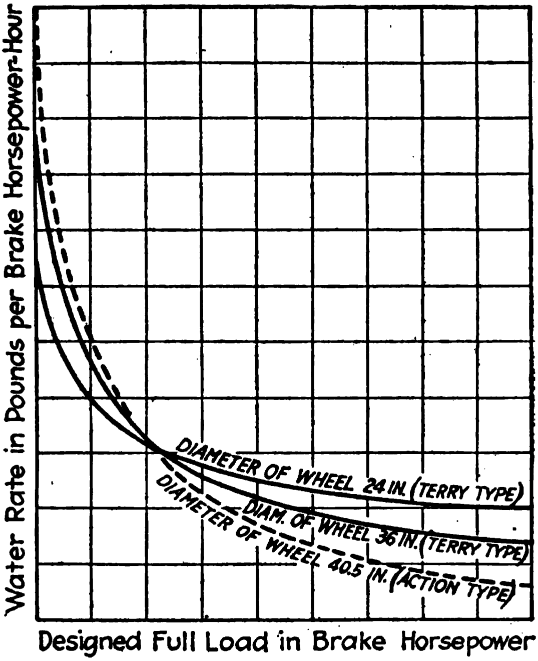


FIG. 1.—RELATIVE EFFECTS ON WATER RATE OF VARYING DIAMETER OF WHEEL AND DESIGNED FULL LOAD.

The characteristics shown are due to increased windage losses on the larger wheels. All curves are plotted for the same conditions of steam pressure.

operating condensing sufficient to offset the first cost, maintenance, etc., the turbine has been at a disadvantage as compared with the reciprocating engine.

The average thermal efficiency of small turbines is in the neighborhood of 40 per cent. With this efficiency all exhaust steam can be utilized without "blowing off," so that no higher efficiency is required or even desirable.

Forty per cent. efficiency does not represent the highest available in this class of machine, but it is conceded to be about the highest commercial efficiency. The cost of small turbines varies approximately directly as the square of the diameter of the runner, whereas the efficiency is increased approximately only inversely as the square root of the runner diameter, so any saving in steam consumption must be accompanied by a marked increase in first cost.

Above 500 to 600 H.P. the field of the large turbine, where water-rate efficiency is of paramount importance, is approached. These machines operate condensing in the same proportion that the small machines operate noncondensing, and the whole problem of design must be attacked on a totally different fundamental basis.

For powers of, say 150 to 500 H.P., condensing, where high efficiencies were desirable, the reciprocating engine until recently had no serious competitor in the steam turbine. The reason for this is obvious. The reciprocating engine was developed, and the meager demand for small

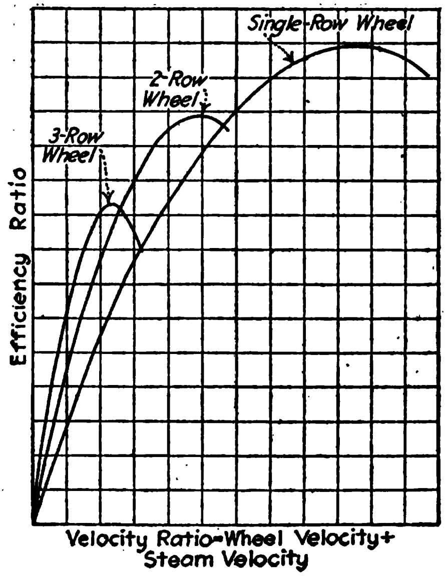


FIG. 2.—RELATIVE EFFICIENCIES OF IMPULSE WHEELS WITH VARIOUS NUMBERS OF ROWS OF BUCKETS.

high-efficiency turbines did not warrant the manufacture of special machines, and furthermore, the customer would not pay the price that would have to be charged.

Conditions have changed rapidly during the last two years or so, and there is now a big demand for small condensing turbines of high efficiency, both high- and low-pressure, which has led to the development of a third class of machine to meet the requirements of this market. To distinguish this class from the small and the large machines, it is permissible to call it the "intermediate design." This design should have, as far as possible, the simplicity and accessibility of the small machine with an efficiency approaching that obtainable with the larger units.

When Parsons and DeLaval built their first turbines the main trouble was not with the turbine itself, but with the "other end," or the driven unit. A turbine is of little use by itself, and it was not until it was demonstrated that it had come to stay that generator, pump and blower makers awoke to the fact that they must remodel their apparatus to meet turbine requirements.

Rapid as the turbine development has been, it would have been more so had it not been for the slow development of the "other end." And past events have again repeated themselves in the field of the "intermediate design." This machine would not have been possible had it not been for the rapid strides that have taken place along the following lines: (a) The manufacture at reasonable cost of high-tensile steel for turbine wheels; (b) the increase in permissible speed of generators, blowers, etc.; and (c) the introduction on a commercial basis of the speed-reducing gear.

In "Power" of October 28, 1913, a brief description was given of the return-flow condensing steam turbine that had just been developed by the Terry Steam Turbine Co. for this so-called "intermediate field."

On the completion of tests of the first machine, several modifications and improvements naturally suggested themselves and are incorporated in the latest designs. For turbines of small power the latest tests show some remarkably good efficiencies, as will be seen by the details of the tests published herewith.

One of the main changes in design has been to carry the principle of the horizontally-divided case to the last extreme. In the original return-flow machine the casing was divided horizontally, but the center diaphragms and the center-diaphragm glands were not, whereas in the machine shown all diaphragms and diaphragm and end glands are thus split, reducing disassembling and assembling time to a minimum. See Fig. 5, right.

In the larger frames another important change has been made. The high-pressure wheel of the Terry type which was incorporated in the first machine has been superseded by a two-row multi-velocity type of wheel running at a high peripheral speed. Extensive experiments have shown that, up to certain peripheral speeds and certain powers with a given thermal drop, the Terry type of bucket is well adapted, but beyond this range, the two-row bladed wheel has the advantage. See Fig. 5.

There are several factors entering into the correct design of a wheel of this type other than the actual or theoretical blade efficiency, which make this problem interesting and more complex than one would suppose from a superficial study of the subject on a purely blade-efficiency basis. Disc friction, the power transmitted or rated power of the machine, commercial considerations regarding first cost (which controls the selling price), are all big factors independent of any blade-efficiency theory.

The return-flow turbine is designed so that the pressure in the first stage will be about 2 to 5 pounds above the atmosphere. With ordinary steam pressures of, say 150 pounds, and allowing two impulses, the peripheral velocity of the buckets must be about 636 feet per second for best efficiency. At 3,600 r.p.m. this calls for a pitch diameter of 40½ inches. For three reversals the diameter of wheel would be in the neighborhood of 24 to 26 inches.

Fig. 1 shows the relative efficiencies of three types of wheels and the important relation that skin friction and windage bear to the overall efficiency. A "two-velocity stage" wheel is more efficient from a blade-efficiency standpoint than a "three- or four-velocity stage" (see Fig. 2), yet the friction created by the increased diameter is far more undesirable than one would at first imagine, and the advantage gained by augmented blade efficiency is more than offset by the added losses in other directions.

The various formulas of Stodola, Lewicki, Odell and others, for skin friction of discs, show clearly how big a factor this can be, and while these formulas are somewhat vague and indefinite regarding certain conditions that have to be taken into account, they all agree that this friction loss varies as the second to the 2.5 power of the diameter of the wheel and nearly as the cube of the peripheral velocity. Again, these formulas do not take into account the windage of the blading, which is obviously greater in a bladed wheel than on a bucket wheel.

In reverting to the bladed type of wheel in the High-pressure end, instead of the semicircular-type bucket, it is interesting to note in passing that the first turbine experimented with by E. C. Terry in 1893 was on this principle. Fig. 3 shows the wheels and guide blades of this early machine, the patent number of which is 508,190.

The Terry type of machine is, primarily and essentially a noncondensing turbine. Its simplicity and consequent unlikelihood of derangement make it an ideal machine for the duties that it is called upon to perform. Within certain limits of speed, vacuum, etc., the two- or three-stage Terry combination makes an equally good condensing machine, having a thermal efficiency as high as that of the single-stage noncondensing design; but

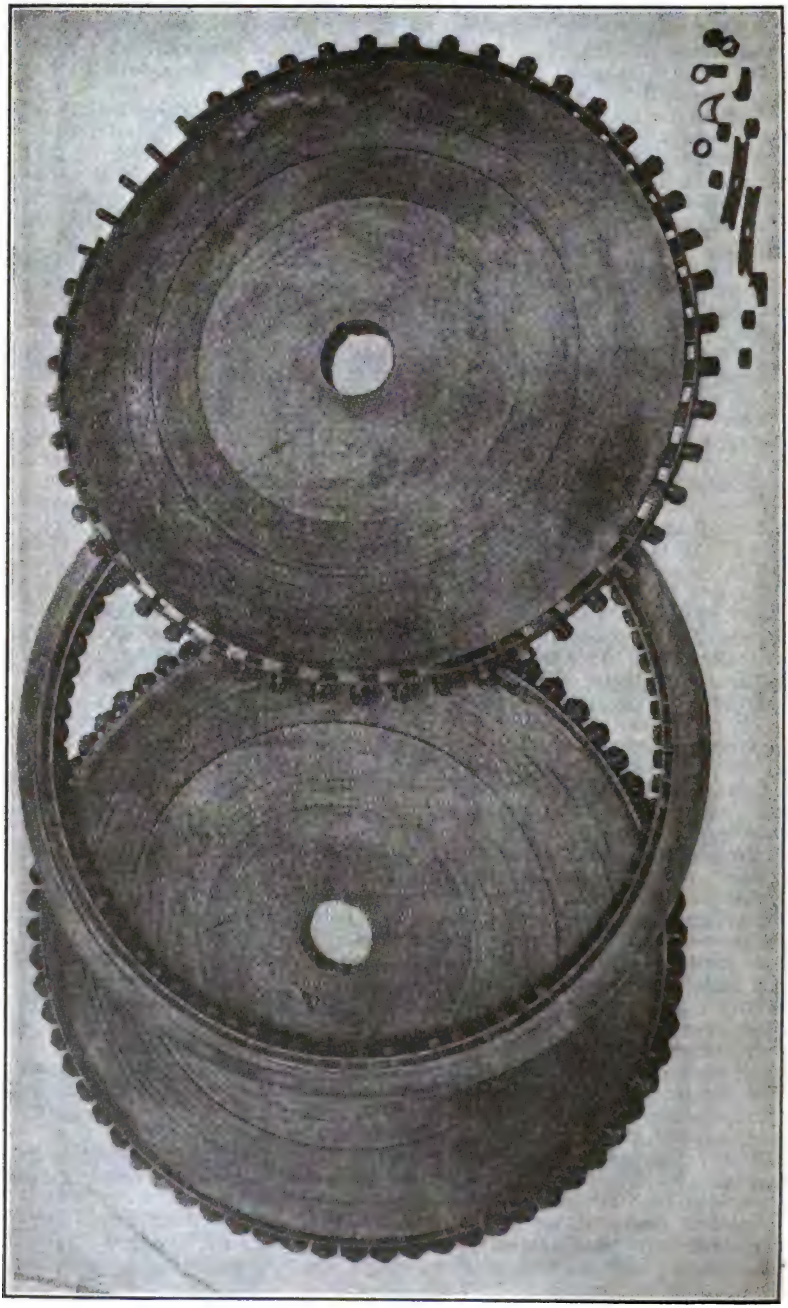


Fig. 3.—Wheels and Guide Vanes of Early Terry Turbine.

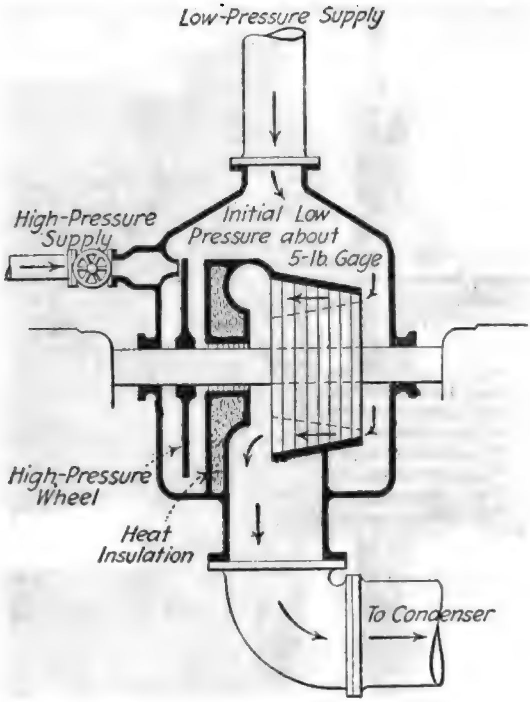


FIG. 4.—DIAGRAMMATIC SECTION OF RETURN-FLOW TURBINE.

when confronted with the necessity for high speeds, high vacuum and larger powers, the wheel becomes impracticable at the low-pressure end owing to its inability to handle a large volume of steam to the best advantage. In the later machines, therefore, the low-pressure wheel has been replaced by a series of single velocity-stage impulse wheels. That practically all authorities agree that this type of wheel for low-pressure work forms the ultimate turbine element is evidenced by the fact that it is being adopted by practically all turbine builders of both large and small machines, with the one exception of the builders of the reaction, or Parsons, type; and that this type of machine is not adaptable to small powers is evidenced by the fact that the builders themselves resort to the impulse principle in their smaller designs.

Again, the "composite design" of velocity staging in the high-pressure end and pressure staging in the low-pressure must be the last word in turbine development if latest designs of practically all the turbine builders both here and in Europe are any criterion.

The obvious advantage of high vacuum in a turbine, particularly in a low-pressure turbine, with the difficulty of designing, building or keeping glands vacuum tight without the necessity of a water seal with its attendant piping and subsequent sediment troubles, led to the departure from the orthodox straight-flow principle to the return-flow design for the elimination of this long-standing bugbear in turbine work. This question of gland leakage often results in trouble between the turbine and the condenser makers when trying to meet guarantees, while the customer

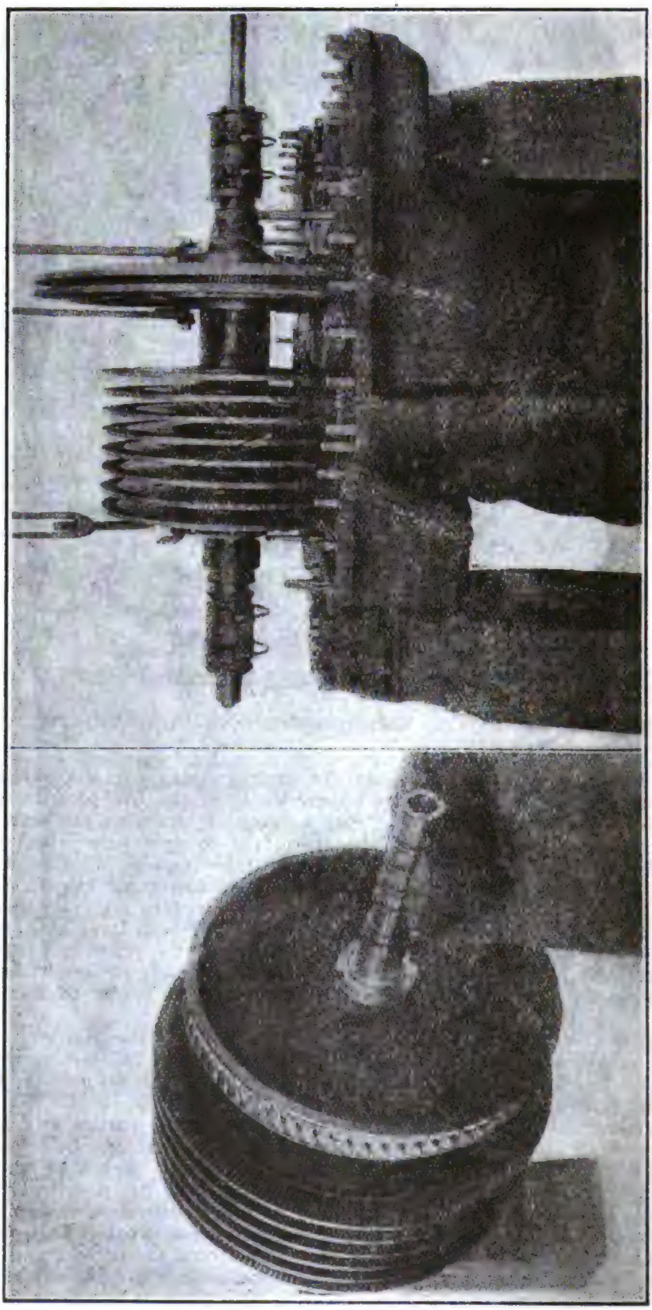


FIG. 5.—ROTORS OF OLD AND NEW DESIGNS OF RETURN-FLOW TURBINES.
Old design of rotor having Terry high-pressure bucket wheel. Rotor of new machine having two rows of impulse blading on the high-pressure wheel.

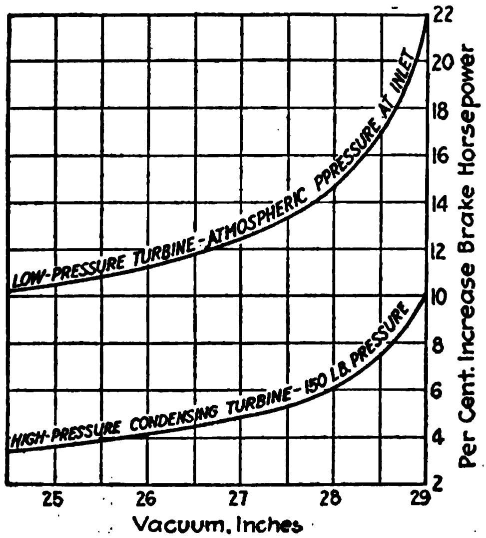


FIG. 6.—PERCENTAGE INCREASE IN POWER AVAILABLE (THEORETICAL) PER INCH VACUUM WITH CONDENSING TURBINES.

looks on and sees the machine run at a lower vacuum than called for and pays the coal bill anyway.

It is often advocated that with a steam seal on the glands and a little steam blowing outward into the engine room there cannot be any air leaking into the turbine. This contention is wrong, as has been demonstrated many times. It often happens that there is a counter current going on, air traveling along one part of the gland and steam passing out of the glands in the opposite direction, this condition being the hardest possible phenomenon to detect. Fig. 4 shows diagrammatically the construction of the return-flow turbine with the low-pressure end in a complete envelope of steam above atmospheric pressure, eliminating the possibility of air leaking into the casing. The rotor and the lower half of the casing of the return-flow machine are shown in Fig. 5.

One other important feature in connection with the arrangement of glands on the return-flow turbine is that no supplementary steam supply is necessary for sealing them when under full load, and even at light loads any steam that finds its way through must pass through the low-pressure end of the turbine, thereby doing work, whereas with the ordinary type of steam-sealed glands all the steam which does manage to escape goes directly to the condenser without doing any further work. That this auxiliary steam supply can amount to quite a factor is evidenced by various tests that have been made. Of course, when a machine is new the glands are tight, so that the leakage during this period is imperceptible, but after setting the machine for commercial operation or if it has been in operation for some time, it is hard to know without repeated tests what this steam leakage amounts to. In big machines this is never, however, a serious amount, but in small ones it can be quite a big percentage of the total steam used.

A careful analytical study of the performance of labyrinth glands was

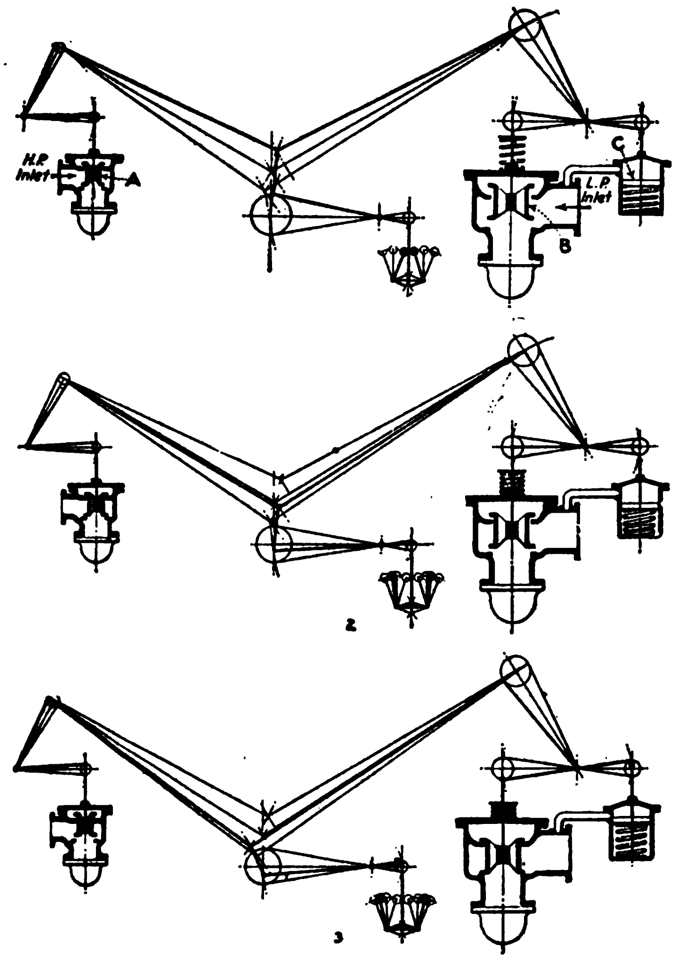


FIG. 7.—DIAGRAMS OF LINK MOTION TERRY MIXED-PRESSURE GOVERNOR CONTROL (RATEAU SYSTEM).

1. Position when starting turbine; high and low-pressure valves open. Piston C forced down by ample supply of low-pressure steam.
2. Speed regulation: High-pressure valve closed. Turbine running on ample supply of low-pressure steam.
3. Pressure regulation: Low-pressure supply stopped; piston lifted by spring, closing low-pressure valve and opening high-pressure valve. Governor is always free to close both valves if load is suddenly taken off the turbine.

made and published by H. M. Martin, and the formula derived from his experiments is given in his book on steam turbines, as follows:

$$W = 68\,A\sqrt{\frac{P_1\left(1 - \frac{1}{x^2}\right)}{V_1\,(N + \log_e x)}}$$

where

W = Weight discharged in pounds per second;
A = Area in square feet available for flow;
P_1 = Initial absolute pressure in pounds per square inch;
V_1 = Initial specific volume of the steam;
N = Number of points at which the steam is wire drawn;
$x = \frac{P_1}{P_2}$, where P_2 denotes the absolute pressure on final discharge from the last ring of the packing.

This formula checks up fairly closely with actual tests made by the writer on a 2,000-kw. machine having a mean labyrinth diameter of 8 inches and 12 elements or restrictions.

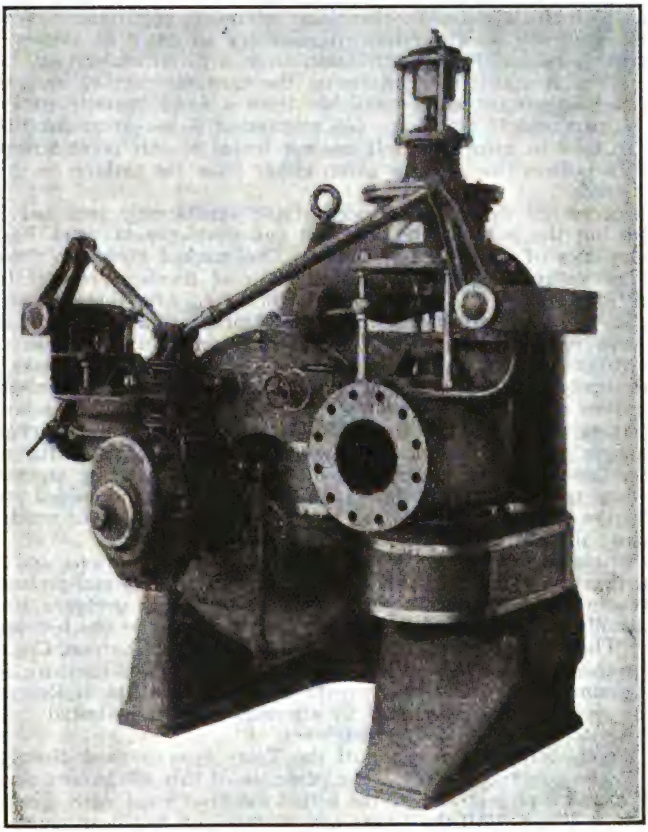


FIG. 8.—TERRY RETURN-FLOW TURBINE WITH RATEAU MIXED-PRESSURE CONTROL MECHANISM.

From the formula it will be seen that the amount of steam passed by a labyrinth gland is directly proportional to the diameter of the glands. This diameter of gland does not follow any relation to the output of the turbine, and it will be seen that the smaller the machine the larger the

percentage of steam that will be passed by the gland; so, as mentioned above, while the amount of steam passed by a labyrinth gland in large machines can be an insignificant factor, it is obvious that in the small machines it can be a serious item. For instance, the figures mentioned above in connection with the 2,000-kw. machine show the total steam passed as 177 pounds per hour. On the basis of 15 pounds per kw. this would give a percentage loss due to the glands of 0.6 per cent., whereas reducing this quantity in the ratio of the diameter of the glands, namely, 8 inches to, say 5 inches on a 200-kw. machine, the gland leakage would be reduced to 111 pounds, but the percentage of the total steam consumption would be increased to 2.5 per cent., the latter based on a water rate for the smaller machines of 22 pounds per kw.-hour.

Fig. 6 shows the theoretical saving per inch of vacuum in a straight high-pressure condensing and a low-pressure turbine. In the low-pressure machine the vacuum is, therefore, of much importance. We must not look upon this as a question of efficiency so much as a question of how much horsepower one can obtain from a given amount of exhaust steam. Then it means that increasing the vacuum from 27 inches to 28 inches the amount of power available from a fixed quantity of exhaust steam is increased 15 per cent. No precaution is too great for the purchaser to take to insure himself against losses at this point irrespective of any guarantees that may be given either from the turbine or the condenser builder.

The success of the low-pressure turbine intelligently installed is undisputed, but the bulk of the research and development work has been along the lines of the larger machines. The marked saving in these mechines has naturally led to the introduction of the low-pressure turbine in small plants such as breweries, ice plants, etc., with just as successful results as with the larger units. Low-pressure turbines of 50 H.P. and more have recently been installed, and many more installations are in course of construction. With the exception of a few isolated cases where a fixed supply of exhaust steam can be depended upon indefinitely, the low-pressure turbine has given place to the mixed-pressure machine, the latter having the advantage that should anything happen to the engine or other source of low-pressure steam supply, the full power of the turbine can be obtained when operated with high- or mixed-pressure steam. The return-flow turbine is particularly applicable to low- and mixed-pressure work, as the effect of vacuum in a machine of this kind is of much more importance than in a high-pressure condensing machine.

For satisfactory operation under mixed-pressure conditions and where the low-pressure supply is liable to decrease or fail, a special arrangement of governor mechanism is designed, so that the high-pressure steam is automatically admitted to make up for any deficiency in the low-pressure supply. The system employed by the Terry Steam Turbine Co. on its mixed-pressure turbines is what is known as the mixed-pressure Rateau control, manufactured under license from the Rateau Steam Regenerator Co., the design being modified to eliminate the complicated oil-relay mechanism necessary on larger machines.

The direct-connected governor of the Terry type is used directly connected to the governor valves. The principle of this mechanism is shown in Fig. 7, and a photograph of the actual machine fitted with this control is shown in Fig. 8. All the levers are mounted on ball bearings to eliminate friction as far as possible. The governor running at high speed has more power than the usual low-speed geared governor and the mechanism itself being balanced by counterweights as shown in Fig. 8, the governor is relieved of all external loads other than to operate the balanced valves.

In all turbine practice, both in high- and low-pressure machines, the question of pipe connections and the elimination of stresses on the turbine is well known to be a serious problem. This is particularly so with the

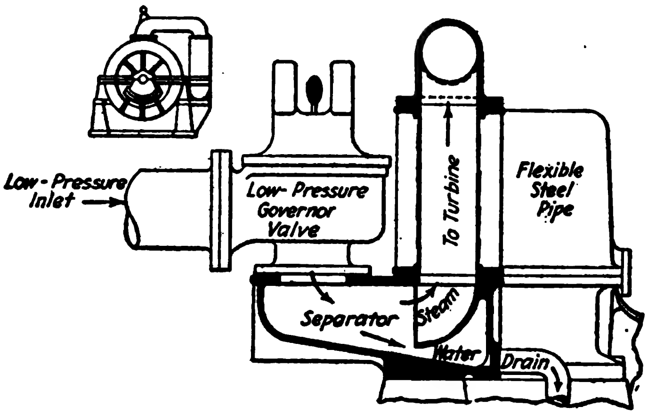


FIG. 9.—ARRANGEMENT OF LOW-PRESSURE STEAM INLET ON RETURN-FLOW TURBINE.

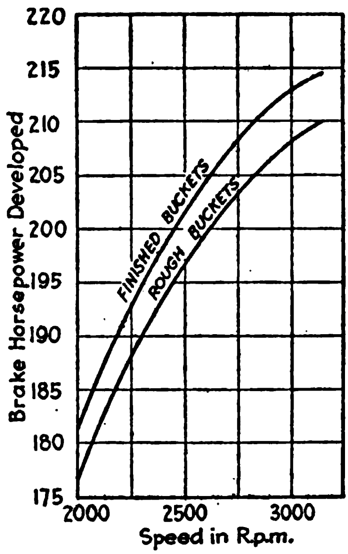


FIG. 10.—TESTS OF IMPULSE WHEELS, SHOWING EFFECT OF FINISHING BLADES.

big piping necessary for low-pressure machines, and it is again of further importance in the smaller units, which, on account of their compact size and light weight are susceptible to distortion from outside stresses. The common practice is to bolt the low-pressure valve and piping directly to the turbine casing. This entails considerable risk of pulling the turbine and, consequently, the whole unit out of line, causing vibrations and generally unsatisfactory running. To eliminate this in the return-flow turbine the low-pressure steam supply is not rigidly connected with the tur-

PERFORMANCES OF TERRY RETURN-FLOW TURBINES.

Turbine number.	Initial steam pressure before governing valves, pounds.	Vacuum exhaust, ins., Hg. referred to 30-inch barometer.	Load in B.hp.	Load in kw.	Speed in r.p.m.	Initial steam condition, degrees F. superheat or quality.	Total steam per hour, actual, pounds.	Total steam per hour corrected to dry steam, pounds.	Water rate per kw.-hour actual pounds.	Water rate per B.hp.-hour actual, pounds.	Water per B.hp.-hour corrected to dry steam, pounds.	Thermal efficiency ratio	Water rate on which efficiency ratio is based.	Test.
1383	128.8	25.64	368.1	...	3,640	10.6	6,297.8	6,370	...	17.10	17.29	0.535	Dry steam.	A
1383	128.8	28.00	441.3	...	3,640	Dry.	...	6,370	...	...	14.43	0.561	Dry steam.	B
1383	128.8	24.16	254.5	...	3,640	10.0	5,016.0	5,069	...	19.7	19.90	0.491	Dry steam.	C
1383	128.8	28.00	343.0	...	3,640	Dry.	...	5,069	...	...	14.77	0.548	Dry steam.	D
1383	7.03	25.60	307.0	...	3,600	85.0	8,086.0	...	...	26.35	...	0.606	Actual.	E
1383	5.20	23.75	193.0	...	2,000	89.0	7,076.0	...	...	36.68	...	0.529	Actual.	F
1383	2.46	24.75	205.5	...	3,000	93.7	6,063.0	...	...	29.48	...	0.649	Actual.	G
1383	8.25	24.35	307.5	...	3,300	76.5	8,340.0	...	...	27.10	...	0.639	Actual.	H
1874	1.75	25.30	...	106.4	2,500	12.0	6,220.0	...	58.5	38.4	...	0.521	Actual.	I
1874	100.4	28.8	...	98.8	2,462	0.981	3,440.0	3,310	34.85	22.85	22.00	0.443	Dry steam.	J
1874	100.0	0.0	...	98.2	2,525	12.5	6,065.0	6,142	61.8	40.58	41.10	0.414	Dry steam.	K
1750	125.5	25.25	416.9	...	3,600	26.9	6,400.0	6,575	...	15.35	15.78	0.597	Dry steam.	L
1750	125.5	28.00	490.9	...	3,600	Dry.	...	6,575	...	...	13.39	0.609	Dry steam.	M
1750	125.0	26.57	321.7	...	3,600	21.0	5,133.0	5,245	...	16.30	15.96	0.558	Dry steam.	N
1750	125.0	27.36	233.4	...	3,600	25.0	3,774.0	3,870	...	16.17	16.59	0.512	Dry steam.	O
1750	125.5	25.25	419.6	...	3,200	26.9	6,400.0	6,575	...	15.26	15.67	0.604	Dry steam.	P
1750	125.5	25.25	407.6	...	2,800	26.9	6,400.0	6,575	...	15.71	16.13	0.587	Dry steam.	Q
1750	125.5	25.25	384.9	...	2,400	26.9	6,400.0	6,575	...	16.63	17.08	0.554	Dry steam.	R
1750	0.25	26.01	67.0	...	500	110.0	6,874.0	...	...	102.6	...	0.175	Actual.	S
1750	0.25	26.01	167.5	...	1,500	110.0	6,874.0	...	...	41.04	...	0.438	Actual.	T
1750	0.25	26.01	237.6	...	3,500	110.0	6,874.0	...	...	28.94	...	0.620	Actual.	U
1750	0.25	26.01	227.8	...	4,000	110.0	6,874.0	...	...	30.19	...	0.595	Actual.	V
2044	124.7	26.49	343.0	...	3,600	27.4	5,165.0	5,305	...	15.06	15.47	0.578	Dry steam.	W
2044	125.0	26.49	295.0	...	3,600	Dry.	...	4,665	...	...	15.82	0.566	Dry steam.	X
2044	125.0	28.00	325.0	...	3,600	Dry.	...	4,665	...	...	14.36	0.568	Dry steam.	Y
2044	125.0	26.49	240.0	...	3,600	Dry.	...	3,990	...	...	16.62	0.539	Dry steam.	Z
2044	125.0	26.49	180.0	...	3,600	Dry.	...	3,260	...	...	18.10	0.495	Dry steam.	AA

bine proper, but is bolted to an entrainer or separator which in itself is bolted rigidly to the bedplate (see Fig. 9). From this entrainer the steam is led vertically through a flexible steel pipe which leads to the top of the turbine casing. In this way no outside stresses due to the heavy low-pressure steam-supply piping are thrown on the turbine itself. Furthermore, by the introduction of this additional entrainer, drier steam is obtainable in the turbine than would be the case were the inlet piping connected directly to the turbine proper.

Throughout the design of the machine special attention was paid to obtaining the best possible water rate consistent with a compact and reliable mechanical design, and the figures given in the table of some recent tests show some remarkably high efficiencies for turbines of such small power.

The last degree of efficiency can be obtained only by a careful study of the correct areas through the blade passages and by special attention to the finish of the blading in the wheels to eliminate friction and eddies. By the adoption of drawn material for the blading, true areas can be obtained. This machine, as built, conforms closely to calculations, as has been evidenced repeatedly by the careful observation of pressure drops through the various stages, these falling almost exactly in line with the calculated drops. The adoption of polished wheels further enhances the efficiency.

Some interesting experiments were carried out to determine the effect of rough and finished blades and wheels. Fig. 10 shows two curves, one with rough drop-forged blades, the other with blades polished to a true section and with the wheel polished.—" Power."

GEARED MARINE TURBINES.

At a recent meeting of the Birkenhead and Liverpool Foremen's Mutual Benefit Society, Mr. J. Hamilton Gibson, the engineering manager of Messrs. Cammell, Laird and Co., gave some interesting data as to the type and dimensions of the teeth used in recent important installations of geared turbines. It might be thought, he said, that very heavy teeth would be required in cases in which 15,000 horsepower were being transmitted per shaft. That, however, was not the case, as for all large powers the standard pitch was only 0.583 inch, so that the teeth were actually less than 5-16 inch thick. That fine pitch led to smooth running, but the wheel and pinion must be held accurately in position and the pinion thoroughly supported. The teeth were all involutes, with a short addendum and large radii at the roots. By adopting the involute form all gears could be cut with the same standard hob, and, further, the oil clearance could be adjusted by opening or closing the distance between the wheel centers by a few thousandths of an inch. With involute teeth such an adjustment did not in any way effect the proper meshing of the teeth. In ships propelled by geared turbines the propeller thrust had to be taken by a thrust-block, and Mr. Gibson said that when it was proposed to take on single collars the screw thrusts of the *Buenos Aires* and *Monte Video,* which developed 5,000 horsepower, many engineers predicted disaster; but the Michel system had proved perfectly successful, the engineers reporting that during the whole run to Buenos Aires the "thrust-bearings ran quite cool, with the chill not off."—" Engineering."

MARINE ELECTRICAL ENGINEERING.*

THE REMARKABLE DEVELOPMENT IN THIS BRANCH OF ENGINEERING.—INSTALLATION ON THE UNITED STATES COLLIER "JUPITER."

By H. A. Horner.

Although it is to be expected that electrical marine applications should be rapid, the developments in this branch of engineering in the last five years are remarkable. Minor applications which were in the experimental stage three years ago are now regular requirements for the more severe and larger service. Such equipments as electric bilge pumps, electric steering gear, etc., though they were utilized on certain foreign ships-of-the-line years ago, held no place in our practice until recently. It is a long span from Jacobi's electrically-propelled boat of 1838 to the United States fleet collier *Jupiter* of 1913; but the gap of years contains progress in the art that will stabilize the application.

GENERAL INSTALLATION.

This survey of advancement must be brief. Following the usage of the time, 80 volts direct current was first adopted. The battleships *Kearsarge* and *Kentucky* building in 1899 were equipped with a 3-wire system using 80 and 160 volts. This precedent led to the standard voltage of 125 volts adopted in 1902. Last year the voltage was raised to 230 volts direct current.

This increase of potential, caused by the expansion of power applications, is disadvantageous to the lighting system and searchlights. The development of the 230-volt tungsten lamp in this country is not such as to insure its adoption at the present time, and searchlight lamps require an arc voltage of 50 to 55. These conditions have brought about the consideration of a system employing either 3-wire generators or 2-wire generators and balancers. Thus there is provided 230 volts for the power system and 120 volts for lighting and searchlights. A standard 3-wire generator has not been adopted and the proposition, therefore, must be classified experimental.

Early in 1913 the standard-conduit installation which had for some years been undergoing modifications and experimentations gave place to lead-covered steel armored conductors. This required new designs and methods for its proper installation, entailing changes in all the appliances, fittings, and fixtures, throughout the equipment. It has led to a great reduction in grounds, it reduces the space and weight required for installation, it allows for further expansion, and incidentally presents a less obtrusive appearance.

Much improvement has been sought in the use of special insulating compositions to supersede porcelain. Various types of asbestos lumber, ebony asbestos wood, and like patented preparations are now generally required. Control panels, interior fittings, etc., are constructed of such materials and at the present time seem to give satisfaction. These experiments have led to the requirement of some such material for the main switchboards superseding slate for this purpose. As this is a recent requirement no switchboards of large size have been manufactured and the questions involved rest in the hands of the designer.

As the tungsten lamp has improved in land practice it now becomes acceptable in marine work. As stated above the voltage (120) is still maintained the same because of the tungsten lamp. Vibrations in certain

*Presented before Section D of the American Association for the Advancement of Science at the Philadelphia meeting, December 30–31, 1914. Mr. Horner is chairman, sections committee, American Institute of Electrical Engineers, Philadelphia, Pa.

parts of the vessel, naturally in the vicinity of the engine spaces, is severe, and carbon lamps cannot be entirely eliminated. The introduction of the tungsten lamps has awakened a renewed interest in the principles of good illumination, and experiments are now under way with prismatic globes, special reflectors, etc.

Many of the above points, and perhaps all combined, have brought improved methods of light distribution and the manner of controlling light. This in turn has caused more economical methods of running feeders to centers of distribution and so produced an installation that is far superior to the old methods in cost of upkeep and reliability of service. Incidentally it has tended to simplicity which materially affects the cost both first and last and points to a hoped for standard.

The old automatic magnet control of the arc has now been replaced by the motor-operated lamp. Experiments with different compositions of carbon so as to produce greater intrinsic brightness at the source is in progress at the present time. The foreign manufacturer is employing an impregnated carbon which permits a higher potential at the arc, 70 to 75 volts instead of 50 to 55 volts. A lamp using the alcohol spray is under consideration. This lamp is of interest due to the increased illumination. A small positive carbon is used and high temperature of the arc is effected by the use of the alcohol spray. It is understood that impregnated carbons are also employed. It is not known whether any attempts have been made to utilize high-power gas-filled tungsten lamps for this purpose, but it is supposed that such a field would be promising. The pendulum swings between the advantages of remote electrically-controlled searchlights and hand or mechanically-controlled lights; but at this time the electrically-controlled light is not in favor.

POWER SYSTEM.

The steady improvement and continued use of contractors for the contol of motors has increased the use of automatic control for a varied number of applications. Dynamic braking, which quickly brings the motor armature to rest, is asserting its certain importance in many equipments.

Electricity is now solely depended upon for weighing the anchor and, although the results of this application for all conditions of service are not known, it is reasonable to believe that the application will warrant its continuance.

Experimental work for the last five or six years has brought electrical steering gear into what appears to be a permanent requirement. The older systems tried did not give satisfactory results and it was not until the development of the contactor that experimental work was again resumed. There are now two systems designed which will be shortly tried in service. One is based on the use of two motors controlled by contactors; the other based on one motor controlled by a motor generator on the Ward-Leonard system. Any type of electrical system is today paralleled by the old steam system, but the indications are that the electric gear will supplant the steam.

The introduction of oil fued instead of coal has changed the electric forced-draft fan to a steam turbine-driven fan, due in part to the fact that high pressure and low volume of air is required and also to the fact that the exhaust from these fans aids the heating of the boiler-feed water.

The main drainage pumps are electrically driven, but the application to other main engine auxiliaries has not increased generally. There is a movement toward rotary steam pumps instead of the old reciprocating pumps, which doubtless will lead in the end to the electric motor.

INTERIOR COMMUNICATION.

The old types of incandescent-lamp instruments have almost entirely disappeared and their place taken by the direct-current motor-operated instrument. The communicating systems as a whole have increased

greatly in extent. No absolute measure of this expansion can be given, but it is about 60 per cent.

One of the most important and interesting developments is that of the gyroscopic compass. Since 1852, when Foucault in his laboratory studied the physical action of the gyro and laid down its laws, scientists and inventors endeavored to produce for it some commercial usefulness. It is not the purpose of this paper to describe or narrate the history of this application. Those who are interested will do well to read H. C. Ford's paper entitled "The Electrically-Driven Gyroscope in Marine Work," read before the American Institute of Electrical Engineers at Detroit, June 23, 1914. It answers the intention of these notes of record that recent improvements in the essential point of quickly damping the oscillations has advanced the practical adaptability of this instrument. A demonstration of this improvement was given by Elmer A. Sperry before the 300th meeting of the American Institute of Electrical Engineers, held in Philadelphia, last October.

The improvements in wireless telegraph apparatus have been rapid and important. Most of these advances have been recorded elsewhere and need no further comment. In general the lines of improvement tend to the use of a continuous wave in preference to the undamped wave. Much success has been achieved by the quench-spark system. Matters of practical installation are more carefully looked after, and the value of wireless as a safety factor has not been exaggerated by the press.

Perhaps the most important advance in the protection of vessels occurred in the early part of this year. For many years and by many minds the transmission of sound signals through water was given special consideration. Water as a medium of signalling has many more desirable characteristics than air. The submarine bell attached to the buoy, the lightship or the shore could emit signals such that a vessel equipped with water-tank and receiver telephones could evade those places in which there was danger and so be protected in foggy or thick weather. But there remained no under-water means by which a vessel could itself communicate either with the shore or with another vessel. The submarine telegraph oscillator is the work of Prof. R. A. Fessenden, and provides for under-water inter-communication by the telegraph or telephone. So much interest surrounds this apparatus that it seems desirable to quote from an abstract of R. F. Blake's paper read before the American Institute of Electrical Engineers at Philadelphia, Oct. 12, 1914, "The apparatus consists of an oscillating electric-motor generator which has a strong electro-magnet surrounding a central core on which is an alternating-current winding. This copper tube is attached to a large diaphragm. When the alternating current passes through the core winding it induces a current in the copper tube, which, being free to move, vibrates back and forth, thus setting the diaphragm in vibration. This oscillator can also be used as a receiver." From this brief description it can be seen that a most important field is now covered by this apparatus.

ELECTRIC DRIVE.

Since Davenport, the Vermont blacksmith, exhibited his electric motor in London many attempts have been made to apply electricity to the propulsion of vessels. Many will recall the electric launches at the Chicago World's Fair in 1893. Abroad some light-draught river craft were built and so equipped. A few years ago two electrically-driven fire boats for the city of Chicago were successfully operated. A vessel built in England to ply in the Welland canal was equipped with electric motors and Diesel oil-engine generators. However the first practical application to a sea-going vessel is the United States fleet collier *Jupiter*.

The keel of the *Jupiter* was laid at the navy yard, Mare Island, Cal., on Oct. 18, 1911. The *Jupiter* was placed in commission April 7, 1913. The vessel was designed for a speed of 14 knots, developing 5,500 S.H.P.

with a load displacement of 19,300 tons at a draught of 27 feet 8½ inches. Her performance showed a speed of 14.99 knots developing 7,151.9 S.H.P. with a displacement of 19,452 tons at a draught of 27 feet 7½ inches. She has been in service over a year and a half, during which time she has had two trial trips, performed the regular functions of a collier, steamed 14,000 miles, and successfully arrived at the Philadelphia navy yard on a continuous trip from San Francisco, stopping only in the Panama canal in order to view the operation of this great engineering project. During this service only two repairs have been made to the propelling apparatus. One of these could have happened to any type of engine; the other was of such minor importance that it need not be mentioned.

SUCCESS OF "JUPITER."

The success of the *Jupiter* is many fold. The fuel economy is shown to be 25 per cent. better than the best of light vessels equipped with other methods of propulsion. The propellers are exceedingly efficient, giving the same speed as sister ships with a reduction of 300 to 800 horsepower. The maneuvering qualities of the vessel are markedly superior due to the rapidity of reversal and also to the fact that full power is available for backing. The ship is about 542 feet long and about 65 feet in beam and would be extremely unwieldly if it were not possible to aid the rudder by means of the propelling motors. There are many more advantages, but the all-important matter is well summed up in the conclusion of a paper by Lieut. S. M. Robinson, U. S. N., read before the Society of Naval Architects and Marine Engineers, Dec. 10, 1914. "After all, the greatest test of the satisfactory working of any machinery is whether or not the men who are actually handling and caring for it are pleased with it. If this test applies to the *Jupiter's* machinery it certainly is an unqualified success. In particular is this true if the matter is referred to the coal passers in the fireroom who have to handle much less coal than do the men on sister ships. The ship can make her contract speed of 14 knots without using forced draft at all."

The success of the *Jupiter* has led the United States Government to extend this application, and electric propulsion is now authorized for the battleship *California*, now building in the New York navy yard. This equipment naturally will be an advancement over that of the *Jupiter*. With this development many more advantages may be expected, as electricity provides ready means for the accurate measurement and determination of propulsion factors. As the application grows many problems of the naval architect and marine engineer will be reopened, doubtless to the betterment of water transportation. There remains to the shipbuilding art today many questions that are settled upon theory because facilities are not given to practically record the conditions of performance. This obstruction will be greatly removed by the application of electricity.

PRINCIPLE OF ELECTRIC PROPULSION.

The first announcement of the principles of electric propulsion of naval vessels may be found in a paper read by W. L. R. Emmet before the Society of Naval Architects and Marine Engineers, Nov. 18, 1909. It is to Mr. Emmet that credit should be given for the introduction of what will doubtless prove to be the best and safest means of ship propulsion; to say nothing of the increased possibilities for the advancement of the art of ship design.

The last half decade is replete with progress in the marine applications of electricity. It is noteworthy that these applications have followed land development in the electric field. From an auxiliary of little importance except lighting, electrical applications of power have grown until now we find electricity entering into the main design of the vessel. The marine

engineer is willing to admit that an electric motor will drive a ship, but imagine his astonishment when he finds in the future to what extent the full effect of this application will lead.—"The Marine Review."

ELECTRIC SHIP PROPULSION.

Before a joint meeting of the Western Society of Engineers and the American Institute of Electric Engineers, W. L. R. Emmet, of the General Electric Co., gave an interesting talk on the above subject on the evening of April 26. The development of the high-speed turbine paved the way for electric ship propulsion. Its application in this field had been long foreseen. Mr. Curtis had worked for two or three years on the problem, and since 1900 Mr. Emmet had spent much of his time on the turbine. About six years ago he had first approached the Navy with a view to equipping battleships for electric drive, but at about the same time the question of reduction gearing had been brought to the front and the Navy had been impressed to the extent that the collier *Neptune* was equipped with turbines and reducing gears. The excellent results obtained aroused interest in the general question of reducing the speed between the turbine and the propeller, and as a result Mr. Emmet secured the contract to equip the *Jupiter* electrically. During the two years this ship has been in service it has made a wonderful record. Results 20 per cent. better than from any ship afloat have been obtained, and the equipment is as good as new. The turbines run regularly on a water rate of 11 pounds per shaft horsepower hour, which may be compared to 14 pounds, the best obtainable from a triple-expansion-engine-driven vessel. Naturally, electric propulsion gained in favor, and about a year ago it began to be thought of seriously for battleships. As the advantages of the electric drive increase with the power required, Mr. Emmet had been particularly anxious to equip a battleship, and only within the last few days the contract for the *California* had been closed. An estimate on the cost of installing electric drive showed that a saving of $160,000 would be effected over the cost of the turbine equiment that had been previously planned. In these large powers all sorts of complications arise when the turbines drive the propellers directly or through reduction gearing. With the latter the power must be divided up between a large number of units, as there is a limit to the size and capacity of individual gears beyond which it would not be safe to pass. In ships where the turbines drive the propellers directly there must be a compromise in speed. The turns made by the propellers are much too high, and the turbine runs at about a tenth of the speed it ought to have to give the best results. Besides, there is great complication of piping for high- and low-pressure turbines, and as the pressure in some of this piping is below the atmosphere, air leaks are liable to develop and lower the efficiency by reducing the vacuum. On the other hand, with the modern electric drive the loss cannot exceed 8 per cent. The apparatus is designed so that practically a constant water rate is maintained for all loads.

In the *Lusitania,* with a speed of 180 r.p.m. the propeller efficiency is 62 per cent. The turbines for the *California* will have a speed of 2,200 r.p.m. and deliver to the generator 75 per cent. of the available energy in the steam. In the latter the turbines will be simple, compact machines, while those of the *Lusitania* are enormous. By dropping the propeller speed of the *Pennsylvania* from 222 to 160 r.p.m., the efficiency would be increased 8 per cent., which would just counterbalance the loss by electric propulsion. Comparing the present equipment with an electrically propelled *Pennsylvania,* the efficiencies would bear a ratio of about 63 to 73 per cent.

Investigating the possibilities of reduction gearing held back electric propulsion. Parsons had condemned the latter and favored gears. Reduction gearing has proven successful on small ships running at moderate speeds. As the speed of the vessel increases, however, the ratio of reduction between turbine and propeller speeds becomes greater and the gear problem is more difficult. The General Electric Co. had become interested in gearing and developed a system which was installed on three freighters equipped with turbines. These gears may be applied to cases where electric propulsion is barred, but in the favorable cases the speaker could not imagine any arrangement of gears which would be anywhere near as good as electric drive.

Electric propulsion is to have a wide field of application. The company had recently figured on two large Russian cruisers and on a number for our own Navy. Mr. Emmet stated that he could reëquip the *Lusitania* and save $150,000 per year in the cost of coal. Electric drive for liners so far exceeds engines that the equipment would pay for itself in one or two years.

Slides were thrown on the screen showing the 20,000-ton collier *Jupiter* and its power-plant equipment. At 15 knots the vessel requires 7,000 horsepower. The generator is of simple and rugged construction and is not restricted as to voltage or frequency. It has a capacity very little greater than required by the motors, so that even a short circuit would not result in much injury. The motors are of the three-phase induction type, the stator having bar windings and the rotor a definite wound design provided with external resistance to be used when reversing. The governor is designed much like a tachometer with a system of fulcrums which can be moved in and out and varied through a wide range of speed.

For the *California* each turbine will have a maximum capacity of 18,000 shaft horsepower and on maximum load will require 170,000 pounds of steam per hour. The vessel has a displacement of 30,000 tons and a maximum speed of 22 knots, and yet each of the two turbines driving it is only 14 feet long. The motors are 12 feet in diameter by 11 feet wide. Consequently, the entire equipment occupies comparatively little space, and the first impression would be that it was designed for a tugboat or at least a vessel much smaller than the *California*. Even the auxiliaries will be electric driven, and the only steam piping entering the engine room will be the two leads for the main turbines.

The two turbines will develop a maximum of 36,000 horsepower, which is required to force the vessel to 22 knots. At 14 knots 7,000 horsepower is required. Performance charts showed that the water rates will remain practically constant over a wide range of speed. At 14 and 21 knots it was 10¾ pounds per shaft horsepower-hour, and for the range in speed between these two points it remained between 10 and 11 pounds. At a speed of 15 knots, 28½ inches of vacuum, no superheat and 190-pound gage pressure, the *Jupiter* showed a perfomance of 11 pounds per shaft horsepower-hour. These figures are exceptional and can be obtained only when both the turbine and the propeller are running at their most efficient speeds. By diminishing the excitation with the speed the efficiency is maintained and at the same time the torque is not reduced beyond that which is required. It is simply a case of diminishing the excitation until the propellers are turned at the right speed with the minimum amount of steam. One of the big problems is reversing, but it has been met by using high excitation while the change in direction is taking place.

With the reduction gear the great problem has been to equally distribute the load over the entire face of the gear. With a rigid gear most of the load is applied near the ends of the teeth. In the General Electric design this difficulty has been overcome by a gear made up of separate discs which will give sidewise and distribute the load over the surface.

A number of charts comparing the relative economy of engine-driven vessels, geared turbines and electric propulsion showed the following water rates per shaft horsepower-hour. For the *Vespasian*, with triple-

expansion engines, the water rate was 19 pounds; with geared turbines, 16 pounds; and with electric drive, 12.7 pounds. The *Cairngowan,* with triple-expansion engines, developed a shaft horsepower-hour on 17.3 pounds of steam, and the *Cairnross*, a sister ship with geared turbines, on 14 pounds. It was estimated that either vessel equipped with electric drive would develop a shaft horsepower-hour on 11.77 pounds of steam. The above figures tend to prove the assertion made by Mr. Emmet that electric drive over triple-expansion engines will reduce the water rate about one-third.

In the discussion it was brought out that as induction motors cannot run above synchronous speed, the propellers cannot race. Even in a heavy sea, with the propellers entirely out of water there is no vibration or any indication of a change in speed. As to the proper fields for reduction gearing and electric drive, Mr. Emmet made the general statement that in all ships requiring above 15,000 to 20,000 horsepower, gearing would make a poor comparison. In vessels requiring 10,000 horsepower and less and running at a low speed, reduction gearing would perhaps make the best showing. The field for electric drive is in large merchant ships and all battleships with the exception of torpedo boats and destroyers, where restrictions in weight prohibit its use.—"Power."

TESTING LUBRICATING OILS.*

SYNOPSIS.—An enumeration of the various tests to determine the qualities of an oil. Results obtained are only approximate.

A brief description of a desirable lubricant is that its viscosity should be the least possible which will allow it to stay in place and do the work. Summarizing the commonly desirable characteristics, they are:

1. The oil should possess cohesion.
2. It should possess the maximum possible adhesion.
3. It should be as far as possible unchangeable.
4. It should be commercially free from acid.
5. It should be pure, that is, it should be what it purports to be.

TYPES OF VISCOSIMETER.

The first to be discussed is the viscosity test, which is used to measure the internal friction of the oil, or, as an engineer might put it, the shearing modulus. This test is of value because a lubricant is really used to keep a shaft or journal and its bearing apart. The journal really revolves on a sheet of lubricant, an action which has been described as revolving on the molecules of the lubricant. The ease with which the molecules slide over one another therefore determines, to a certain extent, the friction loss in a bearing.

A fine example of the effect of the viscosity of lubricating oil is furnished by an experience in a certain spinning mill. The mill was operated with power derived from an engine carrying about the maximum load of which it was capable. The lubricant used on the spindles was changed to one which was supposed to be better. It was then found that the engine did not have power enough to drive the machinery in the mill; as a matter of fact, it was unable to start the machine in motion. Examination showed that the only essential difference between the two lubricants was the possession of higher viscosity by the new oil.

The measurement of viscosity of lubricating oils is in a certain sense unsatisfactory, because the results obtained with the different instruments which are available for making this test do not agree among themselves.

*Abstract of paper read by Prof. A. H. Gill before the Detroit Engineering Society, Mar. 19, 1916.

It is therefore customary to state the instrument which was used in determining any quoted viscosity.

One of the most commonly used viscosimeters is the Saybolt instrument. This is of the pipette type, having a tall pipette of rather small diameter immersed in a jacket which may be used for maintaining any desired temperature during the test. The test is made by filling the pipette to a certain point and noting the time of efflux, in seconds, which is taken as the measure of the viscosity of the oil tested. Or the so-called specific viscosity may be determined by dividing the time required for the efflux of the oil by the time required for the efflux of the corresponding volume of water. The Saybolt instrument was developed by the Standard Oil Co. and was used as a standard for many years, and is today.

The instrument most commonly used in Germany, and now coming into rapid use in this country by both the Government and individuals, is known as the Engler viscosimeter. This differs from the Saybolt principally in using a shorter pipette of larger diameter. It is used in the same way, but the specific viscosities as determined by the two instruments do not agree.

None of the commercial viscosimeters really measure the viscosity, because it can be shown that the tube through which a jet is discharged must have a length of from 175 to 200 times the diameter to give a true measure of viscosity. Any of the commercial instruments can, however, be standardized by measuring the times of efflux of standard solutions of cane sugar of glycerin. By such means the readings of these instruments can be interpreted in terms of absolute viscosity in dynes.

Numerous viscosimeters made of glass have been tried, but unfortunately no two glass instruments can be made exactly alike except at prohibitive expense. For this reason, the glass pipette once used as a standard by the Pennsylvania R. R. was abandoned. It should, however, be noted that a glass pipette, calibrated with glycerin as above described, can be used.

The viscosimeters just mentioned are all of the efflux variety, but there are numerous other forms available, and some of them are particularly well adapted for testing the viscosity of certain commercial products other than oils. For instance, the retarding effect exerted on a paddle revolved in a viscous liquid may be used as a measure of the viscosity and is so used with varnishes, glue and paste. Another form consists of a cylinder suspended from a torsion wire. The retarding effect upon this cylinder when swinging back to normal position after a displacement can be used as a measure of viscosity.

It should be particularly noted that the viscosity varies rapidly with the temperature. It is therefore necessary to state the temperature at which any determination was made.

FRICTION TEST.

There is really no satisfactory test of the adhesive quality of a lubricant. It is commonly supposed to be determined by a friction test. This is made by measuring the frictional resistance offered to the rotation of a standard journal in a standard bearing when lubricated with the oil in question. The results obtained depend partly upon the viscosity of the lubricant and partly upon its adhesion. Modern research shows that viscosity tests show about as much as do friction tests, but this is not necessarily so, and must not be assumed to be universally applicable.

GUMMING TEST.

A third test, and one which is of great importance, is known as the gumming test. This is particularly applicable to petroleum oils and is used to indicate the extent to which the oil has been refined. It serves

indirectly to indicate the extent to which the oil may be expected to change due to oxidation when in use. Numerous opportunities have been offered to check the results obtained with this test and results obtained in practice with the same oils, and all of this experience tends to show the great value of the gumming test.

This test is made by putting a small quantity of the oil to be tested in a small glass vessel, such as a cordial glass, and then mixing with it an equal quantity of nitrosulphuric acid. A properly refined oil will show little, if any, change, but a poorly refined oil will be indicated by the separation of large quantities of material of dark color. This color is due to the oxidation of the tarry matter contained in the lubricant. Experience has shown that oils containing large percentages of tar absorb the most oxygen, that is, they are mildly drying oils.

The results obtained by the gumming test agree well with carbon-residue tests made by distilling to dryness in a glass or a fused quartz flask. The carbon-residue test has been found of great assistance in choosing a satisfactory cylinder lubricant for gas engines, as a large amount of carbon means trouble in the engine cylinder. The lowest carbon content mentioned by the author was 0.11 per cent. The oil giving this test showed no tarry matter when tested with nitrosulphuric acid. In general, a gas-engine oil should not contain more than 0.5 per cent. carbon as determined by the carbon-residue test.

FLASH, FIRE AND EVAPORATION TESTS.

When an oil has been found to have satisfactory viscosity and has given satisfactory results in a gumming test, it must next be checked for safety, that is, the flash and fire test must be made. The amount of volatile matter given off at the temperature at which the lubricant is to be used is often of great importance. A case is on record in which a serious mill fire was spread by vapors given off by the lubricant used in the various bearings. The oil used in this mill gave off 25 per cent. of volatile material when raised to 145 deg. F.

It is advisable to include an evaporation test with the flash test of lubricants. The evaporation test is made by exposing about 0.2 gram of oil at a proper temperature and determining the loss by weight in a given time.

The flash test is made by heating the oil slowly in a vessel surrounded by a proper bath and determining the lowest temperature at which a flame passed over the surface will ignite the vapors which are given off.

FREE ACID TEST.

It is generally conceded that lubricants should be practically free from acid, and the so-called free acid test is made to determine the extent of acid content. The mineral oils are agitated with sulphuric acid during the refining process for the purpose of removing tarry materials, and this acid must be practically all removed from the oil before it is put on the market. Oils may become contaminated with acid from another source as well; namely, the animal or vegetable oils which are occasionally mixed with them for the purpose of modifying their characteristics. A content of 0.3 per cent. of acid is generally considered the maximum allowable.

SPECIFIC GRAVITY.

It is often desirable to determine the character of the raw material from which a given lubricant was made. This can be done in the case of oils refined from petroleum by means of the specific-gravity test. Experience has shown that lubricants made from petroleum with an asphaltic base run from 7 to 10 deg. Baumé heavier than similar lubricants made from petroleum with a paraffin base.

In examining oils, it is well to bear in mind that the viscosity is easily increased by the use of a material known as oil pulp or oil thickener. This is really oleate of alumina, and while it brings up the viscosity, it does not give the greasiness expected when that particular viscosity was specified. At ordinary temperatures, a small quantity of this material will greatly raise the viscosity.

COLD TEST.

There is another test, known as the cold test, which is of value in some cases. If an oil is to lubricate a bearing, it must be fluid enough at the temperature of use to readily flow into that bearing. Many ruined bearings and some fires have resulted from the use of an oil which became too viscous to flow under the conditions of use. For such reasons, it is customary to chill samples of oil and to determine the temperatures at which they become too thick to flow readily.

IODINE TEST.

Tests other than those already described are often made on animal and vegetable oils. They are generally made for the purpose of determining whether the oil under test is what it is supposed to be. It is a simple matter to mix different animal and vegetable oils in such a way that they will give a product capable of passing any one or possibly two given tests, but it is impossible to make such a mixture successfully pass all of the tests which would be passed by the pure oil for which the mixture is to serve as a substitute.

The chemist is often at a great disadvantage in testing such mixtures, because there are no exact specific tests for some of the animal and vegetable oils. The presence of some can be determined absolutely, but unfortunately, this is not true of all.

The iodine test, by which is meant the determination of percentage of iodine absorbed by the oil under set conditions, has long been used to indicate the character of vegetable and animal oils present in a sample. At one time it was believed that the so-called iodine number was a constant for any one oil and that this test was therefore perfect. It is now known that this is not true, the iodine number varying with the condition of the material from which the oil was made.

It is a simple matter to determine the presence of petroleum oils in a mixture of oils with animal or vegetable origin. This is done by saponification, which serves to separate the petroleum oil, which does not saponify, from the others which do.

MAUMENE TEST.

There is a comparatively new test, known as the Maumené test, which gives results comparable with those obtained with the iodine test, but is much simpler and therefore more readily performed by the average individual. For this test, 50 grams of oil and 10 c. c. of sulphuric acid are placed in a beaker and slowly stirred with a thermometer. The maximum temperature rise which occurs is noted and used as an indication of the character of the oil.

TESTS ONLY APPROXIMATE.

It should be appreciated by the practical man that the tests of lubricating oils give only approximate results. Thus any one viscosimeter as ordinarily made will give consistent results on the same material at the same temperature, but different instruments of the same type and apparently exactly alike will give results on the same material which vary several per cent. Similarly, large errors are often obtained when using friction machines. With tests otherwise properly conducted, it appears that the

absorption of oil by the metal of the journal and bearing may be sufficient to cause appreciable errors. Tests have shown that it may take several hours to eliminate the effects of the last oil tested so as to get correct results with a given sample.

No rigid directions can be given for the choice of oils for given purposes. It is best to try various lubricants which can be purchased for any given lubricating problem until one is found which gives satisfactory results. This should then be completely tested and the results of the test should be used in writing specifications on the basis of which bids are to be asked. When the problem is handled in this way, the different prices asked for lubricants which will meet the same specifications will often be found most remarkable.—"Power."

"HAND FIRING SOFT COAL UNDER POWER-PLANT BOILERS."

Is the title of a paper just issued by the United States Bureau of Mines, as an aid to the firemen employed in manufacturing establishments throughout the United States.

The paper, which contains descriptions of methods of firing soft coal under power-plant boilers and of methods of handling fire so as to have the least smoke and to get the most heat from the fuel, seeks to meet the needs of the men, many without a technical education, who are employed in small plants of 1,000 to 2,000 horsepower capacity. For this reason the language used is plain and simple, and technical terms have been avoided as far as possible.

The publication under "General Directions on Firing Soft Coal" makes the following statements:

"When burning bituminous coal under power-plant boilers the best results are obtained if the fires are kept level and rather thin. The best thickness of the fires is four to ten inches, depending on the character of the coal and the strength of draft. The coal should be fired in small quantities and at short intervals. The fuel bed should be kept level and in good condition by spreading the fresh coal only over the thin places where the coal tends to burn away and leave the grate bare.

"Leveling or disturbing the fuel bed in any way should be avoided as much as possible; it means more work for the fireman and is apt to cause the formation of troublesome clinker. Furthermore, while the fireman is leveling the fires a large excess of air enters the furnace, and this excess of air impairs good efficiency.

"The ash-pit door should be kept open. A large accumulation of refuse in the ash pit should be avoided, as it may cause an uneven distribution of air under the grate. Whenever a coal shows a tendency to clinker, water should be kept in the ash pit. All regulation of draft should be done with the damper and not with the ash-pit doors.

"In firing, the fireman should place the coal on the thin spots of the fuel bed. Thin and thick spots will occur even with the most careful firing, because the coal never burns at a uniform rate over the entire grate area. In places where the air flows freely through the fuel bed the coal burns faster than in places where the flow of air is less. The cause of this variation in the flow of air through the different parts of the fuel bed may be differences in the size of the coal, accumulations of clinker, or the fusing of the coal to a hard crust. Where the coal burns rapidly, the thin places form.

"Before throwing the fresh coal into the furnace the fireman should take a quick look at the fuel bed and note the thin spots. In a well-kept fire these spots can be usually recognized by the bright hot flame. The thick places have either a sluggish, smoky flame or none at all. In order

to place the coal over the thin places the fireman should take a rather small quantity of coal on his scoop, for it is much easier to place the coal where it is needed with small shovelfuls than with large ones.

"The coal should be placed on the thin places in rather thin layers. If the fireman attempts to fill up the deep hollows in the fuel bed at one firing, the freshly-fired coal may fuse into a hard crust, thus choking the flow of air, causing the fuel to burn slowly and starting new high places. If the high places in the fuel bed are missed on one or two firings the hard crust at the surface will gradually burn through or crack, thus allowing more air to flow through, and the place will get back to its normal condition. Of course, if the high place in the fuel bed is caused by clinker the flow of air will not be free until the clinker is removed with the fire tool. Whatever may be the cause of the high places in the fuel bed, the fireman should remember that they are places where the coal does not burn. There is no use in placing coal on such a place.

"If the firings are too far apart the coal in the thin spots may burn out entirely, allowing a large excess of air to enter the furnace in streams. If those streams of air are not properly mixed with the gases from the coal, only a small percentage of the air is used for combustion, and most of it passes out of the furnace, depriving the boiler of considerable heat. If, for instance, air enters the furnace at atmospheric temperature, say 75 degrees F., and leaves the boiler at about 575 degrees F., it carries away the heat that was absorbed in raising its temperature 500 degrees F. This heat is lost to the boiler. Another loss of heat occurs when holes form in the fuel bed, because pieces of unburned coal fall through the grate when the fireman attempts to cover the holes with fresh coal. Therefore, in order to avoid the formation of holes, firings should be made at short intervals, particularly if, for any reason, the fuel bed must be kept thin."

Copies of Technical Paper 80 may be obtained by addressing the Director of the Bureau of Mines, Washington, D. C.—"The American Marine Engineer."

GALVANIZED IRON AND STEEL.

By John Hamilton Paterson, D.Sc.

The methods which have been devised for the protection of iron from corrosion are legion; and although considerable ingenuity has been shown in some of them, very few have stood the test of time. One of the oldest of these methods, and also one of the most effective, is that of coating the surface of iron with zinc, commonly known as galvanizing. This process was patented by a man named Crawford in 1837.

The original idea in galvanizing iron was merely to coat the iron with a metal which was less susceptible to atmospheric influences; and although this holds good in the case of zinc, there is another and much more important property which makes it a valuable protective agent. This lies in the fact that, if a piece of zinc and a piece of iron are kept in very close contact with each other in an atmosphere or under conditions likely to cause corrosion, the zinc will itself corrode, but will protect the iron in its immediate vicinity and reduce the corrosion on it to a minimum. It is for this reason that boilers have zinc bars fastened inside their water spaces, and that the stern-frames and plates of vessels which have bronze propellers have zinc plates screwed on to them. The phenomena which are responsible for the protective action of zinc are well understood, but are somewhat outside the scope of this article. A piece of tin, on the other hand, when kept in close contact with iron under conditions likely to

cause corrosion, will itself remain perfectly bright, while the iron will disappear rapidly. Iron coated with zinc, therefore, will remain free from corrosion after the surface of the zinc becomes broken or cracked; whereas iron coated with tin, as in tin-plate, will rapidly rust away as soon as the continuity of the tin surface is destroyed.

The galvanizing of iron is effected in three different ways, known respectively as the Hot-galvanizing process, Sherardizing, and Electro-galvanizing. Hot galvanizing is carried out by dipping the articles into a bath of molten zinc, after they have been completely freed from grease, rust or mill scale by treatment with hydrochloric acid. Galvanized sheets are sometimes rolled after this treatment in order to make the zinc adhere more strongly; other than this there is no further treatment necessary. A small percentage of tin is sometimes added to the zinc bath in order to give the finished surface a spangled crystalline appearance. Articles with fine patterns or corrugations on the surface and pipes or bars with threads cut on them cannot be treated in this way, as the threads or corrugations become filled up with zinc. Large and heavy articles, such as ships' plates and corrugated iron sheets, are galvanized exclusively by this process; in fact in England 90 per cent. of the total galvanizing is done by the hot method.

Sherardizing, introduced by S. Cowper-Coles, but first made a commercial success in America within recent years, is a treatment whereby the zinc is volatilized on the hot, clean surface of the iron, where it forms an adherent and sound coating. It is accomplished by packing the articles, after a thorough cleansing, in zinc dust, and heating them in a suitable receptacle to a temperature of about 350 degrees C. The method is only suitable for small articles, but can be successfully applied to threaded articles, such as bolts and nuts, as the coating is thin and even and sharply follows the outline of the article. Sherardized articles have a dull surface, which, however, can be easily polished on a buff.

Electro-galvanizing, although an old process, is only just becoming possible on a commercial scale. It is accomplished by plating the article with zinc by means of an electric current in much the same way as articles of jewelry are plated with gold and silver. It produces a good adherent coat, and like sherardizing, can be used for nuts and bolts, as it does not fill the threads. It suffers from the defect, however, that the zinc coat occasionally begins to corrode without any apparent reason, and it is also an expensive process to apply. Tubes and pipes cannot be coated on the inside by the electrolytic process.

In spite of the fact that a broken surface of zinc is capable of protecting iron from rusting, it is very important that the zinc should be free from cracks and strongly adherent to the iron. The moment the iron surface is exposed the zinc begins to disappear rapidly, and it is not long before so much iron is laid bare that the zinc cannot protect it and then iron corrosion commences. The phenomena which cause the union of iron and zinc to one another have been closely studied and are somewhat as follow.

When carefully cleansed iron is brought into contact with melted zinc or zinc vapor, the two substances unite and form chemical compounds with one another. In the case of iron galvanized by the hot process, examination under the microscope has revealed the existence of four separate layers in the galvanized portion. These are (1) pure iron, (2) a very thin layer of unknown compositions termed the binding alloy, (3) a layer of an iron-zinc compound having the chemical formula $FeZn_3$, and (4) a layer of zinc permeated by crystals having the formula $FeZn_7$. When a galvanized sheet is sharply bent, parting takes place at the surface of the binding alloy; but in blistering and flaking when heated, parting takes place between the third and fourth layers.

With sherardized iron the coating varies with the conditions of the

process and the composition of the zinc powder, from a thin layer of $FeZn_3$, with a more or less distinct layer of binding alloy, to a thick coat of iron-zinc alloys, with a surface layer of pure zinc.

With electro-galvanized iron, coats of very pure zinc or of zinc-iron alloys are obtained, and in all cases a very thin layer of binding alloy.

From the above it will be seen that, whatever the process used, there is a very close union between the zinc and iron when the galvanizing is carried out carefully, and articles properly treated will stand rough handling without serious damage. The successful use of galvanized iron has led, however, to its adoption in a large number of instances where it is worse than useless. Although zinc withstands the action of ordinary atmospheric influences better than iron, it is attacked with the greatest rapidity by slightly acid or alkaline atmospheres. When open fires burning coke are used, the sulphur in the coke becomes converted into sulphuric and sulphurous acids, and if the roof of the building containing the fires be constructed of galvanized iron it will not last more than a few months unless a protective coat of paint is applied. Galvanized-iron buildings in the neighborhood of chemical works will suffer the same rapid deterioration unless protected by paint. It is much cheaper and more satisfactory in cases like these to erect plain iron buildings and protect them with paint only.

Zinc is attacked fairly rapidly by sea water, and although it acts perfectly as a protector to iron when immersed in sea water it is very quickly eaten away when both iron and zinc are exposed. The British Admiralty make a practice of galvanizing the frames and plates of light ships such as torpedo boats and destroyers. The surface of a ship thus constructed is, however, far from being completely covered with zinc. The rivets, of course, are of naked steel and the edges of a great many of the plates have been cut after the plate has been galvanized, thus exposing a raw edge of steel. It is obvious that if such a ship were left in the water without a protective coat of paint the zinc would soon disappear around the rivets and about any exposed edges of steel, and the shell would then be subject to the ordinary dangers of corrosion. It is clear, therefore, that galvanized ships are only ships with a very expensive protective coat applied to them, which must itself be as carefully protected by paint as is the hull of any ordinary merchant vessel. The only obvious advantage in building a ship of galvanized plates is that it allows of longer periods between the times necessary for repainting. If there is any danger of the too rapid corrosion of the very thin plates of which the hull of a destroyer is made, then surely the use of galvanized plates along the air and water line would be an ample precaution, especially at such a time as the present, when the congestion of galvanizing works seriously interferes with the rapid building of destroyers.

Another danger in the use of galvanized iron is illustrated in the constant failure of galvanized-iron steam and water pipes in merchant vessels. It is rarely the case that these pipes are galvanized after they have been bent to the required shape and threaded for screwing together. The pipes are bent hot, and during the heating process the zinc is all burnt off both inside and out for a considerable distance on each side of the bend. This damage is usually concealed by a thin coat of aluminum paint. The threading of the ends of the pipes exposes, of course, the raw iron, and a portion of this projects beyond the union, generally without any attempt at covering it. Even with these exposed places the pipes would be, with a little attention, quite free from danger of corrosion, but the average engineer regards a galvanized pipe as one which can be safely neglected; and it is a very uncommon sight, even with winch steam pipes laid along the deck, to see them with a proper coating of paint.

In the case of wire rope the galvanizing of the strands of the rope has been of the greatest possible service in prolonging the life of the rope.

In old galvanized-wire ropes corrosion usually started in the center of the rope, and was the more insidious as its effects could not be seen from outside. The coating of zinc on the wire, owing to the methods now employed, is usually very perfect and of considerable thickness. There is little fear, therefore, of corrosion taking place in the center of such a rope during its life.

In conclusion, it is to be noted that, while galvanizing is an efficient method of protecting steel and iron, it is open to several objections. One of these is the tendency to put too much faith in its protecting powers, and to leave galvanized structures to take care of themselves without adequate protection for the zinc. Another is that, in putting together buildings or other structures with galvanized iron, it is impossible not to have here and there exposed parts of uncovered iron. Such parts are frequently inadequately protected, and act as centers from which corrosion may start. Finally, it must be remembered that zinc, while resisting ordinary atmospheric influences, is very much more rapidly attacked by abnormal acid and alkaline atmospheres than iron.—"The Shipbuilder."

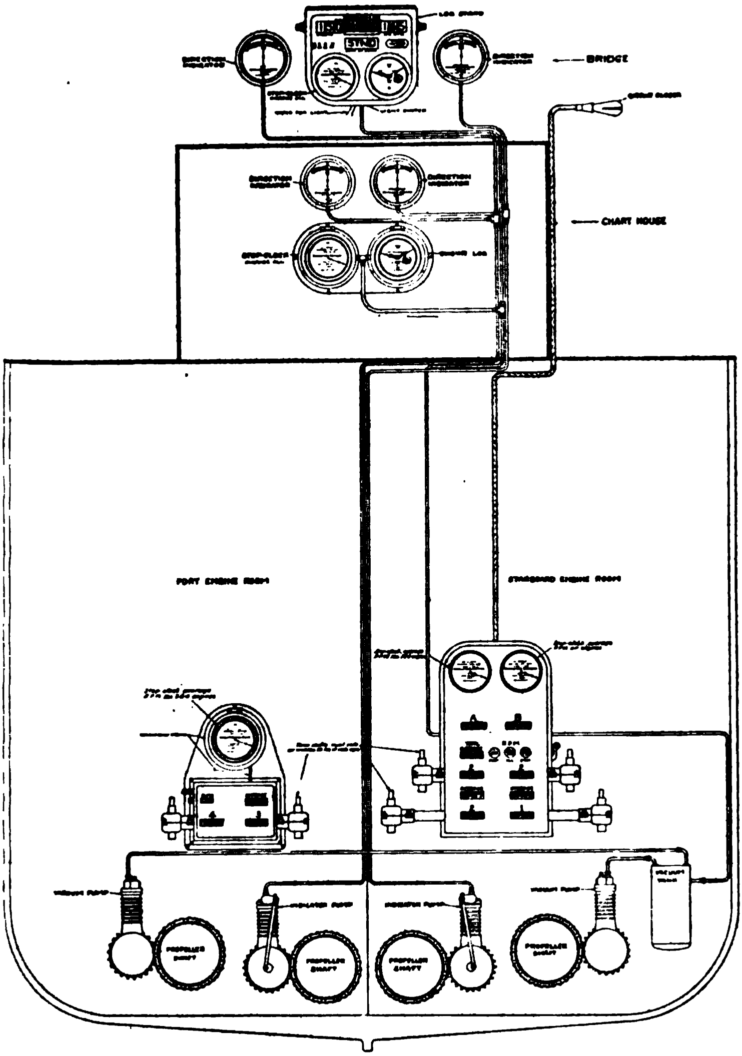
SPEED CONTROL ON DREADNAUGHT *PENNSYLVANIA*.

To promote harmony between the engine room and the bridge, and make it possible to get just that speed from the engines which is necessary for definite purposes, a system of instruments is to be used on the *Pennsylvania* which gives direct and immediate information at all times on the important subject of speeds of the several shafts, together with corresponding speed of ship and other similar matters. The essential parts of this installation, as fitted in the two engine rooms, are shown in Figs. 1 and 2, while Fig. 3 shows the interior of the instrument indicated in Fig. 1. The instruments on bridge and in chart house repeat automatically this same information. The apparatus was designed and built by the Cummings Ship Instrument Works, Boston, Mass.

While the instrument shown in Figs. 1 and 3 is complicated in appearance, it is really simply a combination of what in previous ships has been several independent instruments. The nine revolution counters have been in many cases fitted in three or more separate instruments; while the two stop clocks at the top have usually also been separate from the revolution-counter equipment, and the telltale is also frequently separate. The instrument as a whole, however, has many interesting features.

The four shafts of the *Pennsylvania* are operated from two engine rooms, two shafts in each. The counters from shafts 1 and 2 in the starboard engine room are run by the gears shown at the right of the instrument, and corresponding with the figures given under those shaft numbers. Similarly, shafts 3 and 4 in the port engine room are arranged on the left of the instrument. The revolutions of the shafts are transmitted to the instrument through the "one-way" gears at right and left respectively, these being arranged so that the counters always *add up*, regardless of the direction of rotation of the engine. The counters, therefore, always tell the total number of revolutions made, whether those were ahead or astern, and thus indicate the amount of work done by the engine.

By a combination of gears the indications of shafts 1 and 2 are averaged in the counter shown between those of the two shafts. It will be noted by a comparison with the figures that the number of revolutions indicated in that average is, in fact, a true average between the figure for No. 1 and that for No. 2. The use of positive-drive gearing for this purpose,



CUMMINGS ENGINE LOG SYSTEM INSTALLED ON U. S. S. "PENNSYLVANIA."

as shown in Fig. 3, avoids the errors inseparable from other methods of driving counters. In a similar way the average of the port shafts is given, and then by a combination of these two averages we obtain at once the average for all shafts, as shown just above No. 3 at the left of the instrument in Fig. 1.

The counters A and B are used for trial-trip purposes and for recording the distance run on any given course. This is important when running

on dead reckoning or navigating through a fog, or when for other reasons it becomes necessary to know just how far the ship has been running. It will be noted that the sum of the figures under A and B is just equal to the total average of all shafts. When A is running B is stationary, and vice versa. These are controlled from the bridge by an electric circuit so adjusted that only one can run at a time, and the operation is so instantaneously handled that the records are thoroughly reliable in every way.

The telltale at the bottom, showing which engine is running the faster, is controlled by the starboard average counter and port average counter. When the average revolutions per minute of starboard engines is the same as that of port engines the hand does not move. If one side gains, the hand revolves, indicating by its direction of rotation which side is running faster. The dial around the telltale is graduated in revolutions.

The automatic stop clock in the upper right corner is operated mechanically from the starboard average counter. The mechanism is similar to that on the ordinary stop watch, but for continuous use on ship board a special clock had to be devised which is very much more rugged than any stop watch.

The hand of the clock, started from zero automatically, stops after one hundred revolutions of the shaft have been made. It remains stationary, pointing to the revolutions per minute for a period of 75 revolutions, and is then automatically released and returned to zero. This operation is repeated every 200 revolutions.

As the revolutions governing this clock are taken off from the starboard average revolution counter, this indicates the average number of revolutions per minute for shafts 1 and 2. As installed on shipboard, the speed of ship corresponding to revolutions per minute is placed on the outer edge of the dial, and the engineer at once knows just how fast the vessel is moving through the water. If the engines are running faster than 200 revolutions per minute a reading is given oftener than once a minute. The clocks may be thrown out of gear when not in use.

The operation of the instrument shown in Fig. 2, which is located in the port engine room, is identical with that already described. It shows just what the port engines are doing, in the same way that the starboard instrument shows what the starboard engines are doing. The starboard instrument has the further advantage, however, of showing what *all* the shafts are doing, and of carrying the relaying counter for trial trip and other purposes, as well as the telltale and other details.

The indication of average revolutions per minute by a source independent of the stop clock is sometimes found important. This is given at the upper left corner of the rectangular case in Fig. 2, the knurled knob at the left of the case being pressed in to connect up the gears. The same indications are given on the right side of the large instrument shown in Fig. 1, not only for the starboard average shafts, but for the port average and all shafts. As the cylinders from which these readings are taken are made very compact, and consequently with small figures, magnifying lenses have been introduced to make readings easy.

Figure 4 shows a plan of the entire installation, engine logs, stop clocks and revolution-direction indicators being installed in chart house and on bridge, operated by vacuum and controlled by geared valve shown at middle of left side, Fig. 1. The engine logs show the distance traveled and are used in determining position of ship. They have been found to be the most reliable means for determining position, since they are based on revolutions of propellers. The system has been worked out in such detail that it is quite unnecessary for the navigating officer to telephone to the engine room, or communicate in any other way, to learn what the engines are doing. The information is all before him in the automatic

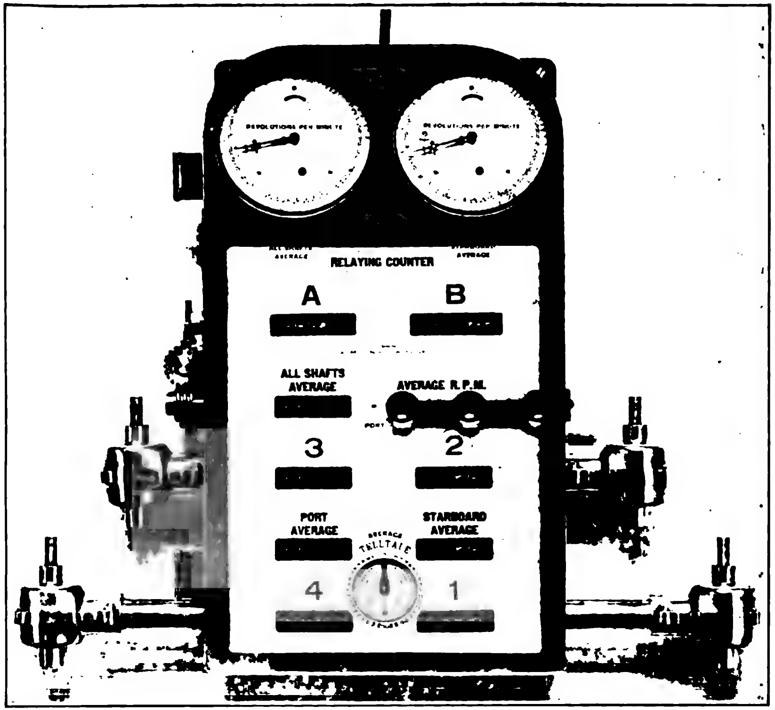

REVOLUTIONS PER MINUTE
REVOLUTIONS PER MINUTE
RELAYING COUNTER
A
B
ALL SHAFTS AVERAGE
AVERAGE R.P.M.
PORT
3
2
PORT AVERAGE
STARBOARD AVERAGE
TELLTALE
4
1

FIG. 1.

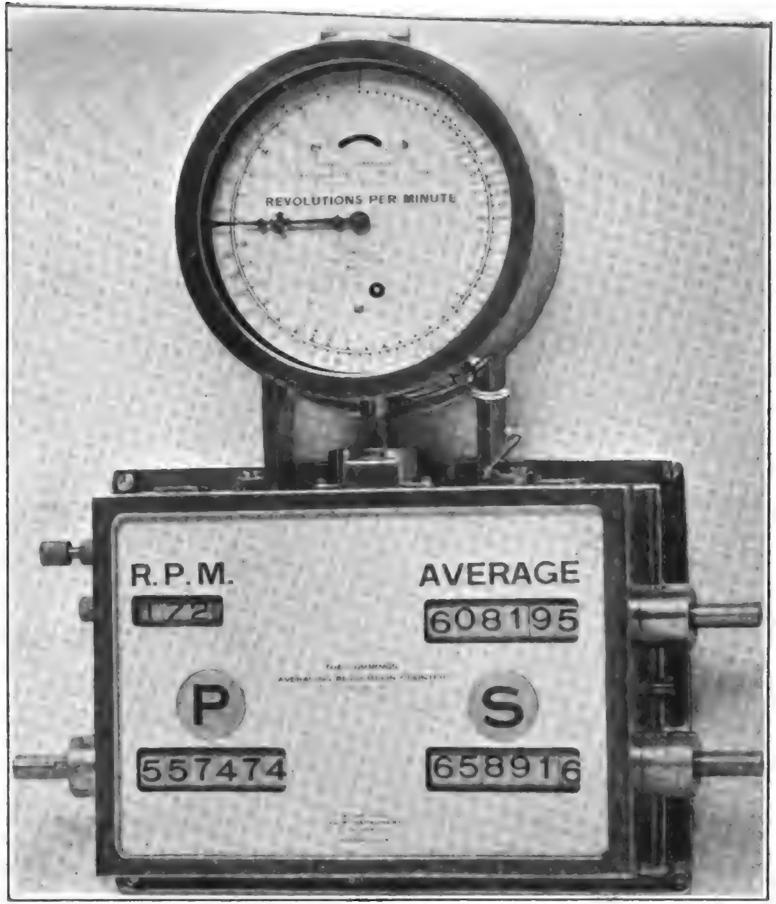

REVOLUTIONS PER MINUTE
R.P.M.
172
AVERAGE
608195
P
S
557474
658916

FIG. 2.

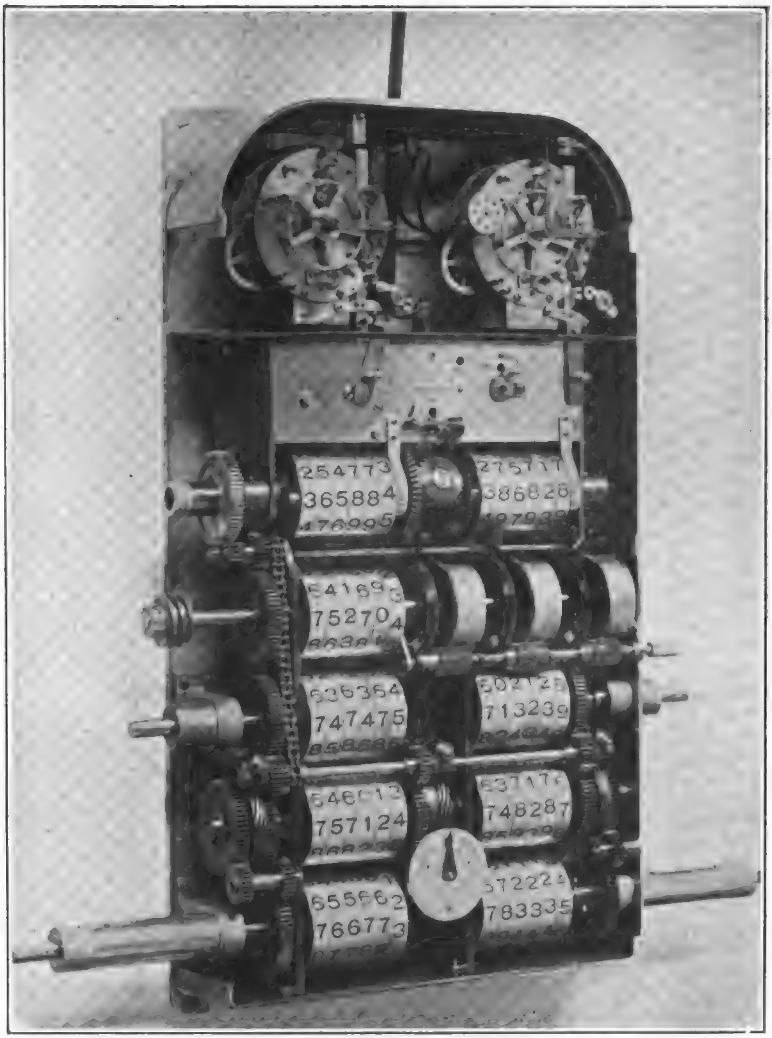


FIG. 3.

instruments—he can see at a glance just what the conditions are, and can take what steps may be necessary to produce the result he desires.

Equipment similar to the *Pennsylvania's* has been fitted on practically every dreadnaught in the United States Navy and on many of the earlier ships. It happens, however, that in the other cases the system has not been carried out to quite the same degree of thoroughness, although the major items of the information are transmitted automatically to the bridge in all cases.—"International Marine Engineering."

PAPERS AT THE INSTITUTION OF NAVAL ARCHITECTS.

A CONTRIBUTION TO THE THEORY OF PROPULSION AND THE SCREW PROPELLER.

By F. W. Lanchester.

The investigations of this paper are based on the idealized conception of a propeller, or "actuator," in which the fluid is assumed to be acted on by forces distributed over a defined surface normal to the axis of motion, representing in sum the magnitude of the total thrust, the said forces being parallel to and opposite in sense to the direction of motion.

Dr. Froude has stated that one-half the acceleration must take place before the propeller and the other half behind it. This view has been attacked by Prof. Henderson and been subsequently further discussed by Dr. Froude.

The present writer reviews the theory from its foundations in order to minimize uncertainty as to the ultimate interpretation of results. While disputing Dr. Froude's conception as to the possibility of transforming the efflux system of flow into a propulsion system, it is admitted that such is strictly analogous. Attention is called to the fact that the Froude actuator must be a plane surface acting normal to the axis of propulsion, and from the analogy of the flow from a re-entrant nozzle which the writer propounds it is suggested that Dr. Froude's results are misleading. The result obtained is that the velocity communicated before traversing the actuator is less than that communicated subsequently. Having established the view that the motion precedent to traversing the actuator contributes to the propulsion reaction, a consideration of the Froude actuator is entered upon independent of the efflux theory. For simplicity sake the actuator is defined by a circle. Roughly speaking, the forces of propulsion will produce a local contraction of the stream before passing through the actuator, followed by an increased contraction which continues until the ultimate contraction of the wake stream is reached, surface discontinuity with the outer flow existing throughout. Theoretical hydrodynamic reasons are given in refutation of the supposed re-expansion of wake stream abaft the actuator; any such phenomenon must be attributable to interdiffusion or mingling between the wake stream and the surrounding fluid.

In introducing a modified hypothesis, it is assumed that the surface of discontinuity between the wake stream and the surrounding fluid degenerates rapidly into a number of vortex rings. Substituting a series of impulses for continuous action, each impulse will give rise to a number of vortex rings which will represent a definite quantity of momentum. A consideration of momentum thus engendered shows that the existence of counter wake does not affect the momentum quantity in analysis. It is claimed that this vortex emission theory directly applies to means of propulsion as at present known.

From consideration of artificial conditions which are regarded as approximating to actual conditions four deductions are reached, namely, that the momentum communicated to the fluid is quantitatively that which appears in the slip stream; the work done on the fluid by the actuator is quantitatively that which ultimately remains as kinetic energy in the slip stream; the kinetic energy in the counter wake system may be considered as a conservative system; both the kinetic energy and momentum imparted to the fluid forward of the actuator contribute to propulsion.

In accordance with the conditions assumed, more than half the acceleration is communicated to the slip stream in front of the propeller, and under the worst conditions the whole acceleration may be ragarded as antecedent.

The author concludes that, if by any mechanical device a continuous stream of vortices could be generated, the actuator will need to operate on fluid moving at a speed half way between the velocity of the general stream and that of the maximum imparted to the vortices; that is to say, Dr. Froude's result is arrived at.

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FURTHER MODEL EXPERIMENTS ON THE RESISTANCE OF MERCANTILE SHIP FORMS, AND THE INFLUENCE OF LENGTH AND PRISMATIC COEFFICIENT ON THE RESISTANCE OF SHIPS.

By J. L. Kent.

Part I of this paper is a continuation of that read before the Institution by Mr. G. S. Baker last year, and gives the results of a further set of model experiments made to determine the effect of shape of area curve on resistance.

Part II gives a general idea of the manner in which the resistance of a ship changes with alteration in form and length. The wave contours at different speeds have been taken for a series of models, and from them an attempt has been made to define some of the conditions which will promote heavy or light wave making.

The results of methodical model experiments in other tanks have been examined and some conclusions drawn from them on the effect of certain changes in form.

Part I.—As the result of experiments made on ten models of identical length, breadth and draught, but varying either in form or fullness, the following findings are arrived at:

Varying bow prismatic coefficient: At low speeds the finest entrance is slightly worse than either the medium or full forms; but for higher speeds it is better than either of them. For higher speeds, the fuller the entrance the worse is the result. At the highest speed to which the full entrance could be adopted, there is a loss of 20 per cent. in power over that for the finest entrance, with a gain of only 7 per cent. in displacement.

Varying stern prismatic coefficient: The results indicate that the fullness of stern has not such an influence on the resistance of ships as fullness in bow has, and although slightly greater resistance may accompany increase in after prismatic coefficient, the larger displacement obtained may outweigh the small disadvantage as regards resistance.

Varying shape of entrance: A hollow-line bow is found to be best for the particular model tried over the range of speeds possible for a ship of the given fullness, but the medium-line bow is never more than 3 per cent. worse. The straight-line bow is much worse for all speeds possible for any ordinary vessel of the said form.

Varying shape of run: The medium-line stern gives better results than either a hollow or a straight line aft.

Part II.—Dr. R. E. Froude has described the "principal wave-initiating operations" of the entrance and run of any ship which give rise to two systems of transverse waves, and he has shown that the coincidence of a bow-wave trough with a stern-wave trough will cause increased wave resistance, whilst a bow-wave crest coinciding with a stern-wave trough causes diminished wave resistance.

The purpose of the present communication is to give the results of experiments made to determine the effect of form of bow on the resistance of the model due to all possible phase difference between the bow and stern-wave systems. Two series of models, each consisting of an entrance and run joined together by different lengths of parallel body, eleven models in each series, have been tested. The difference between the two series was in the shape of the water lines of the entrance, one set having a straight waterline entrance and the other being hollow. In all other respects the models were alike. The bow wave for all models of the same set at the same speed was found to remain unaltered, irrespective of the length of parallel body. As the stern is brought nearer the bow by cutting out the parallel body, the growth of the primary trough of the stern-wave system, as a trough of the bow-wave system passes through it, becomes apparent, and its effect on the residuary resistance is most marked. The humps and hollows of the residuary resistance curves are caused by interference between the bow and stern-wave systems. The humps are evidence of an increased waste of energy, and the wave profiles show that this is due to an increase in the heights of the resultant system of waves formed by the model.

Part III.—To trace the effect on the resistance of a ship of a change in the prismatic coefficient or length, an examination of the methodical series of model experiments published by Messrs. R. E. Froude, D. W. Taylor and G. S. Baker are made. The effect of varying prismatic coefficient for ships of 450 feet length and 10,000 tons displacement from Froude's A series is that an increase in residuary resistance accompanies increase in prismatic. From a comparison of Froude's A and B series, the conclusion is arrived at that for ships having the same displacement, length, prismatic and midsection coefficients, the residuary resistance increases as the ratio $\frac{\text{Breadth}}{\text{Draught}}$ increases. From Taylor's experiments it is borne out that increasing the midship section coefficient, keeping the same prismatic coefficient, reduces residuary resistance for the same length ship.

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LAW OF FATIGUE APPLIED TO CRANKSHAFT FAILURES.

By C. E. Stromeyer.

The author read a paper last year on the "Elasticity and Endurance of Steam Pipes," and in the present one he deals with crankshafts on the same lines. This problem is far more complicated than that associated with steam pipes, and the results arrived at are not precise. Thus, although crankshaft failures may be due to insufficient strength for ordinary working conditions, they may also be due to the shafts being out of line or to the bearings having become overheated and suddenly drenched with cold water.

The sudden cooling of a hot crank pin or journal is of course likely to lead to very serious results, for the cold water has access only to the fillet to which the violent contraction is confined, and as this is professedly the weakest part of a crankshaft, cracks are most likely to occur here. The fact that cracked crankshafts can safely be run for a few months suggests that these cracks are only surface deep and may therefore be produced by the sudden drenching of the hot shaft in cold water. The

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other disturbing factor in this inquiry is the alignment of the shafting. The bending of shafts may be due to a settling down of the engine, to discontinuity in the ship's structure, to unequal loading of the ship or wearing of the brasses or to the slacking back of main bearing bolts. The bending moments in crankshafts are easily calculated if the following assumptions are made, as has been done in the paper. The mean indicated load on each piston was assumed to be constant throughout the stroke, the inertia of the moving masses counteracting the irregularities of the diaphragm. On dealing with crankshafts as though they were continuous —a most laborious process—it was found that the after crank was subjected to weaker bending stresses than the forward one and yet most of the failures occurred in the after crank. It was also found that very slight changes in the alignment increased the stresses, and as it seemed probable that with continuous working of an engine the bearings would wear down in such a manner as to equalize loads, this condition was presumed to be the prevailing one. In other words, both crankshafts were treated as if they were connected together by flexible couplings. Under these assumptions the bending moments of the several fillets were calculated and also the torsion moments.

The resisting moments were more difficult to deal with, for the corners of the crank had to be treated as being curved beams, and the author had in all the cases embraced in the paper recorded the radii of curvature of the fillets. Their influence on the resisting moment of cylindrical sections is proportional to the square root of the radius r of the fillet to the diameter d of the shaft, so that the strength of the curved beam of circular section may with reasonable accuracy be expressed by the formula

$$M = \frac{\pi}{32} \times S \times d^{\frac{5}{2}} r^{\frac{1}{2}},$$

where S is the maximum stress; and for an elliptical section the formula would be

$$M = \frac{\pi}{32} \times S \times b \times a^{\frac{3}{2}} \times r^{\frac{1}{2}},$$

where a and b are the major and minor axes of the ellipse.

The above sketched method of estimating the stresses in the fillets is probably fairly correct because such disturbing influences as present themselves seem to balance each other.

Another point of some importance has to be mentioned. In the forward crank fillets there are practically only bending stresses which act parallel to the axis of the shaft, alternating from a maximum tension to a compression of the same intensity and if the estimated alternating stress is, say, ± 5 tons per square inch, the range of alternating stress would be 10 tons. In the after crank-pin fillet the case is different, for when the after crank is at the top center both the torsion moment and the bending moment are at their maxima; at half stroke the torsion moment has died out, and the bending moment, being now at right angles to its first direction, produces practically no bending stresses in the crank-pin fillet. On reaching the lowest position the shearing stress due to torsion is again a maximum, and there is now a maximum longitudinal compression stress in the fillet, which is instantly changed into a tension stress when steam is admitted below the piston. On compounding these several stresses it will be found that the top position, the major right-angle resultant stress, is inclined, say, 10 degrees to 20 degrees, to one side of the axis of the shaft. It gradually dies out, and reappears as the compression stress inclined to the other side of the axis until suddenly it is changed into a tension stress and reoccupies its original direction. Thus the alternating tension and compression stresses do not act in the same direction and an estimated alternating stress of ± 5 tons per square inch in the after

crank is perhaps equivalent to a range of only 7 tons instead of 10 tons.

The result of the author's investigations may be roughly summarized by stating that if crankshafts are subjected to fewer alternating stresses their endurances can be estimated if the fatigue properties of the material are known; and seeing that modern materials for crankshafts have much higher fatigue enduring properties than was produced by wrought iron, modern crankshafts ought to be relatively much stronger than these. Some allowance should of course be made in the factor of safety for external influences, but it would generally suffice to adopt a factor of 1.5, taking the fatigue limit of the material as a standard.

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A COMPARISON BETWEEN THE RESULTS OF PROPELLER EXPERIMENTS IN AIR AND WATER.

By A. W. Johns.

Some years ago, in a paper read by the writer before this Institution, it was pointed out that, in the case of bodies moving through air at practicable velocities, the alteration of pressure is relatively so small that the compressibility of air—the quality in which it differs so materially from water—can be neglected. The coefficients of resistance for the same body moving in the two media—air and water—would thus be in the ratio of the densities. Theoretically this is not correct, but actually it appears to be sufficiently so for all practical purposes.

The rapid progress in aviation during recent years has been accompanied by the publication of a large amount of information resulting from the experimental work of the various aeronautical laboratories. So far as the writer is aware this information is not familiar to naval architects. It is with the object of emphasizing the importance of a knowledge and study of such results that this paper has been prepared. The performances of propellers are of practical importance, and in this paper the results of experiments with them in air have been compared with those obtained in water by Messrs. Froude and Taylor.

The propellers whose results are dealt with are five in number, all two-bladed. The diameter is in all cases 2 feet and the blade widths range from 6⅝ to 2⅝ inches. The shape of the portion furthest from the boss is elliptical, and that of the portion nearest the root is obtained by drawing tangents from the root of the blade to the ellipse.

The mean pitch ratio of all the propellers is .733. This is the nominal face pitch ratio. The published results, given by the Advisory Committee on Aeronautics, give the thrusts and efficiencies at various slips and various speeds of translation. The slips are calculated from the effective or vertical pitch, as used by Dr. Froude in his 1908 paper. A comparison of these results with the experimental data of Froude and Taylor shows a remarkable similarity in performance when the correction for different densities is taken into account.

It is found, however, that for constant slip, and on speed of translation base, that the thrust does not increase in all cases with the second power of the speed, and for narrow blades the power varies as at a greater power than the square of the speed, whilst for wide blades it varies at a smaller power than the squares. The efficiencies of the five propellers are given, and these are observed to increase with the speed, but as the slip is increased the speed at which maximum efficiency for the particular slip occurs gradually decreases. The increase of efficiency with speed is noteworthy, and may be of some practical importance if the same holds good for water. There is no reason, so far as one can see, why it should not. Mr. Taylor experimented at speeds of 3, 4, 5, 6 and 7 knots, but mentions no differences in the efficiencies as the speed varies.

Except in the matter of effective pitch, the aerial propeller results compare extremely well with those of water propellers. The thrusts are in agreement and can be obtained by dividing by the ratio of the densities of the two media. The efficiencies are in agreement at the lower speeds, and the results generally point to the fact that more experiments on propellers are required to clear up some important points. The increase of efficiency with speed, the departure of the thrust from the law of the square of the speed, the applicability of the law of similitude of Lord Raleigh, and the alteration of the slip at which maximum efficiency occurs as the speed is increased, are all of great practical importance. For this reason it is important and necessary that a careful study should be made of all results as they become available from the various aeronautical laboratories.

There are other points of interest to the naval architect in the results published by these laboratories. The best form of strut and the behavior of rudders are two which can be mentioned.

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THE EFFECT OF BEAM ON THE SPEED OF HYDROPLANES.

By Linton Hope.

This paper gives the results of the author's investigations to discover the fastest hull to take a 250 to 300-B.H.P. motor and only 1,100 pounds' weight. Three distinct types, namely, the multi-step, the single-step and the stepless hydroplanes or skimmers were investigated, and the result of these investigations lead the author to the conclusion that the single-step was much superior to the stepless, while the multi-step came somewhere in between the two. By a process of elimination and classification, the results given by the author were obtained. The various types were first tabulated according to ratios of $\frac{W}{P}$ and $\frac{V}{\sqrt{L}}$, and this eliminated the "wasters." They were then further arranged in order of the $\frac{B}{L}$ ratios, and it became apparent that, other things being equal, beamy boats had a decided advantage in speed, at any rate, when the ratios $\frac{V}{\sqrt{L}}$ and $\frac{L}{\sqrt[3]{L}}$ were high and weather conditions favorable.

The author sought the advice of Dr. R. E. Froude, and submitted his formulae to him. Dr. Froude, although approving of the formula generally, strongly objected to the introduction of beam in any form, on the ground that only one dimensional factor was necessary or desirable, and that the proper proportion of beam for any given conditions would be determined by experience, while its introduction in the formula might lead to a wrong conception of its true value when designing future boats. Admitting this fully, there yet remains the fact that knowledge of the best proportion of beam to length is very slight, so, for analytical purposes only, the author ventured to go against Dr. Froude's advice, and used a constant varied by the ratio $\frac{B}{L}$, with surprising results.

The formula as corrected now reads:

$$\text{B.H.P.} = \frac{WV\left(\frac{V}{\sqrt{L}}\right)^{1.4}}{C}$$

for power, and—

$$V_{(\text{kts})} = \sqrt[2.4]{\frac{\text{P.C.L.}^{7}}{W}}$$

for speed. The coefficient C is obtained by multiplying the constant for each type by the ratio $\frac{B}{L}$; thus $C = 66{,}000\,\frac{B}{L}$ for single-step hydroplanes, while for the stepless type it would be $55{,}000\,\frac{B}{L}$, or thereabout. The type constant 66,000 or 55,000 would again vary according to the ratio $\frac{\text{E.H.P.}}{\text{B.H.P.}}$; but, as this appears to vary only from .47 to .48 in most of the examples, .475 was taken as a mean propulsive coefficient, and fixed constants of 66,000 and 55,000 for the two types.

Although the E.H.P. could not be obtained in many examples, some were obtained, notably the case of Sir John Thornycroft's *Miranda IV*, which has the highest efficiency of any example of a racing hydroplane which the author has come across. *Miranda's* ratio of E.H.P. to B.H.P. is .48, while the next highest (also a Thornycroft design) is .478, and most of the others appear to be about .475, except one which is as low as .47.

Taking .475 as the mean, it was necessary to increase the constant $60{,}000\,\frac{B}{L}$ to $138{,}950\,\frac{B}{L}$ to obtain the approximate E.H.P. necessary for towing experiments or for the floats of hydroplanes. For ascertaining the resistance only, the formula is simplified to

$$R = \frac{W\left(\frac{V}{\sqrt{L}}\right)^{1.4}}{K}$$

the value of the constant K being $425\,\frac{B}{L}$ for the single-step and $355\,\frac{B}{L}$ for stepless, while for the multi-step type it appears to be about $400\,\frac{B}{L}$.

Table I gives tabulated data of the various types. All the thoroughly authenticated figures are printed in heavy type; those which have been obtained from reliable sources, but not fully authenticated, are in italics, and estimated figures are in ordinary type.

The *Flapper* was the first vessel to reach the speed of 40 knots, as the trials were run early in the year 1910. *Flapper*, which was designed, owned, and engined by Mr. M. E. Batting, was similar in type to Sir John Thornycroft's *Miranda III*, but much lighter in proportion, and her ratio of $\frac{W}{P}$ 13.44 pounds per horsepower, has only been beaten by the French competitor for the British International Trophy last year.

The author regrets he was unable to include the *Miranda III* in his table, as her best authenticated speed on the Seine is stated to be 51 knots. Although the following figures are believed to be fairly correct, the author was unable to verify them: Length, 7 m., or 23 feet; beam, 1.6 m., or 5.45 feet; weight, 3,360 pounds; power, 325 B.H.P.; and speed, 51 knots; ratio, $\frac{W}{P} = 10.34$ pounds per horsepower.

Returning to the table, after five columns of data there are three columns of the ratios—

$$\frac{B}{L},\ \frac{W}{P},\ \text{and}\ \frac{V}{\sqrt{L}};$$

then in the next column the type constants—

$$C = 66{,}000\,\frac{B}{L},\ 61{,}800\,\frac{B}{L},\ \text{and}\ 55{,}000\,\frac{B}{L},$$

according to the classes A, B, C, and in the adjoining column the constant c obtained by the formula—

$$c = \frac{WV\left(\frac{V}{\sqrt{L}}\right)^{1.4}}{P}$$

The twelfth column gives the percentage of error between the last two columns, and the others give power and speed obtained by the formula with the percentage of error. It will be seen that, except in the case of two boats in Class B and two in Class D, the error is generally well under 1 per cent. Even in these cases the maximum error is only 1.69 per cent., while the average error of all the examples in Class A is 0.37 per cent., in Class B 0.85 per cent., and in Class C 0.25 per cent.

The power of the table is the brake horsepower of the motor minus 3½ per cent. for loss through the reduction gear.

No. 4 in Class B is also of interest, beyond the fact that she is the widest example obtainable, as she is the only one fitted with an aerial propeller similar to that of an ordinary aeroplane. Her performance is above the average, but it is impossible to say how much is due to hull design and how much to propeller efficiency. The difference, however, does not appear to be sufficient to warrant the claim so frequently put forward by aeronautical experts that aerial propellers are far more efficient than the ordinary marine type, unless the effect of beam has now become detrimental on account of the high ratio of $\frac{B}{L}$ 0.36. It is obvious that the beneficial effect of beam must have some limit, and the author expects that limit to be reached with a $\frac{B}{L}$ ratio of 0.3.

The author illustrates with a diagram the variation of the constant C for each type according to the ratio $\frac{B}{L}$. This diagram also shows some results of tank experiments made by Mr. G. S. Baker with five models of floats for hydro-aeroplanes. Unfortunately, these models are all very heavy in proportion to their length and power, their ratio of—

$$\frac{L}{\sqrt{W}}$$

being 1.05 to 1.36, while the author believes that none of the real hydroplanes have ever obtained satisfactory results with this ratio below 1.7.

In the author's examples the average is 1.83, and runs up as high as 1.93, which amounts to a difference of 74 per cent. between the widest model and the average of the real boats of the same single-step type in Class A.

Not only were Mr. Baker's models much heavier in proportion than the real boats, but their power is also lower in proportion, and his mid-sections are more or less rectangular—a form long since discarded on racing hydroplanes. The proportionate speeds at which Mr. Baker's models were run were considerably below those which have given the best results in the racing hydroplanes, the average ratio of $\frac{V}{\sqrt{L}}$ being only 5.69 in the models, while it is 7.42 in Class A, the most successful type of the real boats.

The most interesting of Mr. Baker's experiments are the two models No. M. 43b and No. M 61, both of the single-step type Class A. The length and form of hull are practically the same in both, but No. M 61 is rather more than twice the beam of No. M 43b. Unfortunately, she is also double the weight, so her ratio of $\frac{L}{\sqrt{W}}$ is far below that of any real

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TABLE OF DATA.

No.	L L.O.A.	B Beam.	W Weight.	P B.H.P.	V Speed.	$\frac{B}{L}$	$\frac{W}{P}$	$\frac{V}{\sqrt{L}}$	[illegible]	[illegible]	Error per cent.	[illegible]	Error per cent.	[illegible]	Error per cent.	No.	Remarks, &c.
Class A.—Single-Step Type.																	
	Ft.	Ft.	Lb.		Knots.						Per cent.		Per cent.	Knots.	Per cent.		
1	[illegible]	6·25	[illegible]	240·00	41·00	[illegible]	20·00	7·2[illegible]	15,301	13,[illegible]	−0·20	[illegible]	−0·[illegible]	41·00	+0·10	1	
2	20·00	6·05	2,000	100·00	28·00	·31[illegible]	27·73	6·[illegible]	14,14[illegible]	14,0[illegible]	−[illegible]	[illegible]	−0·[illegible]	[illegible]	+0·20	2	This boat had the engine and propeller of No. 1.
3	[illegible]	7·50	[illegible]	[illegible]	30·00	[illegible]	[illegible]	7·[illegible]	14,[illegible]	14,406	−0·10	[illegible]	−0·11	[illegible]	+0·06	3	
4	[illegible]	[illegible]	[illegible]	40·00	[illegible]	[illegible]	40·60	5·[illegible]	14,[illegible]	14,[illegible]	+0·16	[illegible]	+0·17	[illegible]	−0·10	4	Sister boat to No. 1.
5	[illegible]	[illegible]	2,000	[illegible]	[illegible]	[illegible]	[illegible]	6·[illegible]	14,[illegible]	14,781	+0·[illegible]	[illegible]	+0·[illegible]	[illegible]	−0·13	5	Geared propeller.
6	[illegible]	[illegible]	15,700	720·00	[illegible]	[illegible]	10·23	7·[illegible]	14,[illegible]	14,[illegible]	−0·17	770·70	−0·17	[illegible]	+0·12	6	Speed only attained once.
7	[illegible]	2·40	672	30·00	41·00	[illegible]	12·44	10·[illegible]	14,[illegible]	14,[illegible]	+0·17	[illegible]	+0·16	[illegible]	+0·16	7	Flapper with plane.
8	[illegible]	[illegible]	2,050	53·00	[illegible]	[illegible]	[illegible]	5·[illegible]	15,[illegible]	13,273	+0·27	[illegible]	+0·27	[illegible]	+0·11	8	Miranda IV.
9	[illegible]	[illegible]	3,[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	6·[illegible]	13,[illegible]	13,[illegible]	+0·[illegible]	[illegible]	+0·[illegible]	[illegible]	−0·[illegible]	9	Ditto with small motor.
10	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7·[illegible]	16,704	16,[illegible]	+0·[illegible]	[illegible]	+0·[illegible]	[illegible]	−0·15	10	
11	[illegible]	[illegible]	[illegible]	70·00	[illegible]	[illegible]	[illegible]	9·[illegible]	17,700	17,700	−0·[illegible]	77·[illegible]	−0·[illegible]	[illegible]	+0·04	11	
$C = 61,[illegible]\frac{B}{L}$																	Note.—Formulæ and errors. All the results have been taken out by slide rule.
Class B.—Multi-Step Type.																	
1	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	12,[illegible]	12,[illegible]	−0·[illegible]	[illegible]	−0·[illegible]	[illegible]	+0·[illegible]	1	One hull with different powers and weights.
2	[illegible]	[illegible]	[illegible]	60·00	[illegible]	[illegible]	[illegible]	[illegible]	12,[illegible]	12,[illegible]	−1·[illegible]	[illegible]	−1·[illegible]	[illegible]	+0·[illegible]	2	
3	[illegible]	[illegible]	2,000	[illegible]	[illegible]	·3000	[illegible]	7·[illegible]	12,[illegible]	12,[illegible]	+1·[illegible]	[illegible]	+1·[illegible]	[illegible]	−0·[illegible]	3	
4	[illegible]	[illegible]	1,000	[illegible]	[illegible]	[illegible]	[illegible]	9·[illegible]	[illegible]	[illegible]	+0·[illegible]	[illegible]	+0·[illegible]	[illegible]	−0·16	4	Aerial propeller.
$C = [illegible]\frac{B}{L}$																	
Class C.—Stepless Type.																	
1	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5·[illegible]	11,13[illegible]	11,100	+0·20	[illegible]	+0·[illegible]	[illegible]	−0·[illegible]	1	B.M.B.C., [illegible]-ft. Class.
2	[illegible]	[illegible]	1,[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5·[illegible]	11,[illegible]	11,[illegible]	−0·[illegible]	[illegible]	−0·[illegible]	[illegible]	+0·15	2	
3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	6·[illegible]	11,107	11,[illegible]	+0·[illegible]	[illegible]	+0·[illegible]	[illegible]	−0·[illegible]	3	
4	[illegible]	[illegible]	676	[illegible]	[illegible]	[illegible]	[illegible]	9·[illegible]	12,[illegible]	12,[illegible]	−0·[illegible]	[illegible]	−0·[illegible]	[illegible]	+0·[illegible]	4	Flapper without plane.
$C = [illegible]\frac{B}{L}$																	
Class D.—Displacement Type.																	
1	[illegible]	[illegible]	4,[illegible]	812·00	[illegible]	·1146	[illegible]	[illegible]	[illegible]	[illegible]	+0·[illegible]	[illegible]	+0·[illegible]	[illegible]	−0·[illegible]	1	Dixie II.
2	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	·1[illegible]	[illegible]	4·[illegible]	[illegible]	[illegible]	+1·[illegible]	[illegible]	+1·[illegible]	[illegible]	−0·[illegible]	2	Legru-Hutchins.
3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	·1[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	+1·[illegible]	[illegible]	+1·[illegible]	[illegible]	−0·[illegible]	3	[illegible] II.

skimmer and in addition she was not run at such a high speed as No. M 43b. If, however, she were run at the same speed of 22 knots, it may be assumed from the other examples that her power would decrease from 19.025 E.H.P. to about 15.0 E.H.P., or, say, 31.6 B.H.P., which would give her a constant by my formula of 18,608. As No. M 43b has a constant of 14,504 at 22 knots, doubling the beam has apparently increased the efficiency by about 21.4 per cent., in spite of the much lower ratio of $\frac{L}{\sqrt{W}}$ and a $\frac{B}{L}$ ratio which is probably well beyond the point of maximum efficiency.

This the author considers all in favor of his theory of a fixed $\frac{B}{L}$ constant, at any rate until $\frac{B}{L}$ reaches .3 or thereabouts; but it does not account for the extraordinary efficiency of Mr. Baker's toboggan-shaped stepless model No. M 98, which attained a speed of 24 knots with only 12.75 E.H.P. If we take her probable power at a speed of 22 knots to correspond with the two previous models, it appears to be about 14 E.H.P., or, say, 29.5 B.H.P., which gives her a constant of 9,893, whereas the formula for the C Class only gives her a constant of 7,895; so that even at 22 knots her constant is 25.6 per cent. better than that of any of the stepless racing hydroplanes, particulars of which are given in this paper.—"Shipbuilding and Shipping Record."

NAVAL VESSELS.

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1915. Apr 1	May 1	Percentage of completion May 1, 1915. Total.	On ship.
	BATTLESHIPS:								
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	91.61	92.70	91.7	90.9
37	*Oklahoma*	New York S. Co.	Reciprocating	2	20.5	94.27	94.97	92.9	92.9
38	*Pennsylvania*	Newport News Co.	Cur. trb. grd. cr.	4	21	61.56	65.27	76 9	72.1
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	36.05	37.97	59.2	55.4
40	*California*	Navy Yard, N. Y.	...	4	21	...	...	...	...
41	*Mississippi*	Newport News Co.	Cur. trb. grd. cr.	4	21	1.57	2.56	17.9	13.3
42	*Idaho*	New York S. Co.	Pars. trb. grd. cr.	4	21	4.18	8.60	25 2	10.0
	DESTROYERS:								
51	*O'Brien*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	97.77	97.77	97.3	97.3
52	*Nicholson*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	96.82	98.70	*	*
53	*Winslow*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	89 69	91.57	92.0	91.8
55	*Cushing*	Fore River S. Co.	Curtis trb. grd. cr.	2	29	95 16	95.97	92.0	91.9
56	*Ericsson*	New York S. Co.	Pars. trb. & rec.	2	29	95.88	96 27	96.3	96.3
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr	2	29.5	57.84	59.72	61 7	57.7
58	*Conyngham*	Wm. Cramp & Sons	Pars. trb. grd cr.	2	29.5	6[illegible].69	66.97	67.5	65.3
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	54.96	63.05	66.3	63.9
60	*Wadsworth*	Bath Iron Works	Pars. trb. gearing.	2	30	91.82	93.90	91.2	91.2
61	*Jacob Jones*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	73.30	78.69	68.0	67.5
62	*Wainwright*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	75.14	78.44	66.9	66.4
63	...	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	29.13	33.45	19.5	12.7
64	...	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	29.13	33.45	18.6	9.8
65	...	Bath Iron Works	Pars. trb. grd. cr.	2	30	4.66	12.06	10.5	3.9
66	...	Bath Iron Works	Pars. trb. grd. cr.	2	30	4.66	12.06	10 5	3 9
67	...	Wm. Cramps & Sons	Pars. trb. grd. cr.	2	29.5	...	12.12	7.3	1.5
68	...	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	1.50	2.00	...	...
	FUEL SHIPS:								
13	*Kanawha*	Navy Yard, Mare Isl'd	Reciprocating	2	14	98.30	98 93	100.0	100.0
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	44.10	53.78	87.3	86.7
	SUBMARINES:								
31	*G*-3	Navy Yard, N. Y.	Diesel-Sulzer	2	14	92.00	92.00	88.6	88.4
40	*L*-1	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96 38	97.40	95.8	95.8
41	*L*-2	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96 38	97.40	97.0	97.0
42	*L*-3	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.87	96.89	90.3	90 3
43	*L*-4	Elec. Boat Co., Quincy.	Diesel-New Lond	2	14	95.61	95.99	90.0	90.0
44	*L*-5	Lake Co., Bridgeport	Diesel-Sulzer	2	14	6.68	7.74	66 8	62.6
45	*L*-6	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	6.46	6 48	59.0	53.6
46	*L*-7	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	6.39	6.48	57 2	51.4
47	*M*-1	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	89.86	90.83	73.3	68.1
48	*L*-8	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	...	...	26 3	24 2
49	*L*-9	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	80.92	85.77	66.0	58.1
50	*L*-10	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	80.92	84.07	64.3	55.3
51	*L*-11	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	66.91	71 16	52.7	42 0
52	...	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	...	...	...	...
53	*N*-1	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	...	...	...
54	*N*-2	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	...	...	...
55	*N*-3	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	...	...	...
56	*N*-4	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	...	14.5	11.4
57	*N*-5	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	...	11.9	8 6
58	*N*-6	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	...	11.2	7 9
59	*N*-7	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	...	11.2	7.9
	SUBMARINE TENDER:								
2	*Bushnell*	Seattle Con. & D. D. Co.	Pars. trb. gearing	1	14	82.50	84.80	88.3	83.5
	DESTROYER TENDER:								
2	*Melville*	New York S. Co.	Pars. trb. gearing	1	15	87.00	89.15	88.0	87 9
	Transport	Navy Yard, Phila.	Reciprocating	2	14	3.61	4.41	13.4	6.3
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	2.87	4.47	13.4	4.4
	PANAMA COLLIERS:								
	Achilles	Md. Steel Co	Reciprocating	2	14	86.84	96 38	96.8	92.9
	Oil Barges 8 and *9*.	Navy Yard, Mare Isl'd	...	...	...	88.70	99.98	...	...

* Delivered April 30, 1915.

BATTLESHIP *PENNSYLVANIA.*

The battleship *Pennsylvania,* launched from the yard of the Newport News Ship Building & Dry Dock Co., Newport News, Va., is the most powerful fighting machine yet added to the fleet of dreadnaughts of the United States navy.

The great advance made in naval design and construction is forcibly demonstrated when a comparison is made of the *Pennsylvania* and the dreadnaught *Delaware,* launched at the same shipyard Feb. 6, 1909. The *Delaware* has a displacement of 20,000 tons, and five turrets with two 12-inch guns each, a total of ten 12-inch turret guns; while the *Pennsylvania* has a displacement of 31,400 tons, and four turrets with three 14-inch guns each, a total of twelve 14-inch turret guns.

The *Pennsylvania* on Feb. 1, 1915, was 67.7 per cent. completed, and the contract date of completion is Feb. 28, 1916.

Some of the principal characteristics of the *Pennsylvania* are as follows:

Length on load waterline, 600 feet; length over all, 608 feet; breadth, extreme, load waterline, 97 feet ½ inch; depth to weather decks, main and upper, respectively, 46 feet and 53 feet 9 inches; mean draught to bottom of keel at trial displacement, 28 feet 10 inches; number of decks, 5, with bridge deck and two platforms; number of compartments, about 400; mean trial displacement, 31,400 tons; speed, 21 knots.

Armament.—Main battery:—Twelve 14-inch, 45-caliber, breech-loading rifles. Four submerged torpedo tubes. Secondary battery:—Twenty-two 5-inch, 51-caliber, rapid-fire guns; four 3-pounder saluting guns; two 1-pounder guns for boats; two 3-inch field pieces, and two 0.30-caliber machine guns.

Complement.—Officers, 65; crew, 863; marine detachment, 74.

Fuel oil capacity, 694,830 gallons,—2,322 tons.

The plans and specifications for the *Pennsylvania,* authorized by act of Congress, approved Aug. 22, 1912, were completed and circular signed by the Assistant Secretary of the Navy Dec. 18, 1912, and issued to bidders upon request after December 20, 1912.

Contract for the *Pennsylvania* was signed with the Newport News Shipbuilding & Dry Dock Co., Newport News, Va., on Feb. 28, 1913, at a price of $7,260,000; to have the bidder's design of Curtis turbines installed, and to be completed within 36 months.

The total cost of completed ship is approximately $13,000,000. The keel was laid Oct. 27, 1913.—"The Marine Review."

U. S. S. *WADSWORTH.*

Destroyer No. 60, the *Wadsworth,* was successfully launched at the Bath Iron Works, May 1, 1915.

The *Wadsworth* is one of a class numbered from 57 to 62, contracts for which were let by the Navy Department in the latter part of 1913. The contract price of the vessel is $884,000 and the principal dimensions are as follows: Length, 315 feet 3 inches; beam, 29 feet 10 inches; mean draught, 9 feet 3 inches; displacement on trial, 1,050 tons. The contract speed is 30 knots.

The peculiarity of this destroyer is that it is the first one in the U. S. Navy to be entirely driven through gearing. The main propelling turbines are of the Parsons reaction type and drive twin screws by means of helical pinions and gears. These propellers are designed for 450 revolutions per minute and to develop a shaft horsepower of 17,500.

This vessel has accommodations for 5 officers and 95 men. There are four 4-inch guns and four torpedo-tube mounts, each carrying twin torpedoes 21 inches in diameter.

U. S. S. *MAUMEE.*

The U. S. S. *Maumee,* launched at the Mare Island Navy Yard, March 3, 1915, is the first large Diesel propelled ship for the Navy. The *Maumee* is a submarine tender, of 5,000 indicated horsepower, twin screw, and designed for 14 knots.

PANAMA COLLIER *ULYSSES.*

The Panama Collier No. 1, *Ulysses,* successfully completed her preliminary trials on April 15, 1915, averaging 14.99 knots per hour on her five high runs, and 14.605 for the 24-hour run. The *Ulysses* was built by the Maryland Steel Company under contract dated April 9, 1914, to be completed in 16 months. Contract speed was 14 knots.

U. S. S. *TUCKER.*

Torpedo-boat destroyer No. 57, the U. S. S. *Tucker,* was successfully launched at the Fore River Ship Building Company on May 4, 1915. The *Tucker's* machinery consists of turbines in combination with cruising turbines and reduction gear. There are two shafts, with one ahead turbine, one backing turbine and one cruising turbine on each shaft.

U. S. S. *CALIFORNIA.*

On April 24, 1915, the award of the propelling machinery for battleship No. 40, the *California,* was made to the General Electric Company. The contract price is $441,000, and besides the main propelling machinery includes main air and circulating pumps, blowers for ventilating motors and generators, and exciters for the main generators. The main propelling machinery will consist of two turbo-generators of about 11,000 kw. each, running at 2,100 r.p.m., and four double squirrel-cage induction motors, one on each shaft, running at 175 r.p.m. The generators will be quarter-phase, two-pole, and have a maximum voltage of 3,000 volts. The main machinery will be manufactured and tested at the works of the General Electric Company and installed by the Navy Yard force at Brooklyn, New York.

FRANCE.

THE LATEST FRENCH DESTROYERS.

Among the latest additions to the French Navy are the four destroyers *Aventurier, Opiniatre, Temeraire* and *Intrepide,* which are the largest vessels of their type now under the French flag. All of these destroyers were built and engined by the "Britany yard" at Nantes. Their main particulars are: Length overall, 283 feet; beam, 28.32 feet; depth, 17 feet; draught, at the stern, 10 feet; displacement, 1,100 tons.

The type of hull is the same as adopted in the previous destroyers *Voltigeur, Fourche,* etc., which have been illustrated and described in previous issues of this journal. This type of design, however, has proved very efficient, allowing the propellers to give their maximum efficiency even in bad weather. Their seaworthiness has been amply proved in actual service during the winter months.

The engines consist of two Rateau turbines, each driving a propeller 7

feet 3 inches diameter and 6 feet 9 inches pitch, working at 650 revolutions per minute. The total horsepower developed is about 20,000, and it is reported that speeds of 32 knots were obtained on the measured-mile trials. Steam is supplied by four water-tube boilers, using coal as fuel, and a fifth fired with oil fuel. The capacity of the coal bunkers is 12,362 cubic feet and of the oil tanks 3,179 cubic feet. The destroyers have an acting radius of about 3,000 miles.

The armament consists of four 4-inch rapid-fire guns, all mounted on the centerline of the vessel, and four 18-inch torpedo tubes.—"International Marine Engineering."

FRENCH BATTLESHIPS *BRETAGNE* AND *PROVENCE*.

The battleships *Bretagne* and *Provence*, laid down on May 1, 1912, and July 22, 1912, respectively, have recently carried out their official trials and now form a part of the Mediterranean fleet of the French navy, where they are the largest battleships in the fleet. The main particulars of these two battleships are: Length, overall, 545 feet; beam, 88 feet; draught, forward, 28 feet; draught, aft, 29 feet 3 inches; displacement, on normal draught, 23.546 tons; area of midship section, 2,752 square feet; brake horsepower, 28,000; contract speed, 20 knots.

On the *Bretagne* steam is supplied at a pressure of 256 pounds per square inch by twenty-four Niclausse boilers of an improved type, having a total heating surface of 65,000 square feet and a total grate area of 2,090 square feet. The funnels on this ship extend to a height of 66 feet above the load waterline. On the *Provence* steam is supplied at the same pressure by eighteen boilers of the Guyot du Temple type, having a total heating surface of 63,000 square feet and a total grate area of 1,500 square feet. This is the first time that boilers of the torpedo-boat type have been used on a ship of this size, and the performance of the *Provence* under severe service conditions will be followed with interest—especially so because some of the vessels of the *Normandie* class will also be fitted with this type of boilers.

In both vessels the main engines are Parsons turbines, arranged, according to the latest practice, on four shafts, without cruising turbines. The engines for the *Bretagne* were supplied by the Mediterranean Works and for the *Provence* by the Loire Works.

The electric plant consists of four generators, each of 200 kilowatts capacity. Forward and aft, near the turrets, there are also two other similar electric generators, making the total power of the generating sets 1,700 horsepower. All of the generators are driven by high-speed reciprocating engines.

Nearly all of the auxiliaries on the vessels are operated by electricity. Except in the engine and condensing rooms, all parts of the ship are ventilated by electrically-driven fans. The ammunition rooms are cooled by Westinghouse-Leblanc refrigerating apparatus.

The *Bretagne* and *Provence* are the first battleships of the French navy to carry an armament consisting of ten 13.5-inch guns all mounted on the centerline of the vessel.

GERMANY.

GERMAN SUBMARINES.

At the meeting of the French Society of Civil Engineers, held in Paris on March 26, M. Laubeuf gave the following data on German submarines: Information from the British Admiralty states that, at the commencement of 1914, Germany had 24 submarines ready and 14 in construction; of

TABLE I.—DIMENSIONS OF GERMAN SUBMARINES.

	U 1.	*U 2* to *U 8.*	*U 9* to *U 12.*	*U 13* to *U 20.*	*U 21* to *U 32.*	*U 33* to *U 38.*
Date of commencement	1903	1906–1907	1908	1909–1910	1911–1912	1913.
Displacement on the surface	185 tons	237 tons	Slightly larger than the *U 2* boats.	450 tons	650 tons	675 tons.
Displacement when submerged	240 tons	300 tons		550 tons	800 tons	835 tons.
Length	128 feet 3 ins.	141 feet 8 ins.		...	213 feet 3 ins.	...
Breadth	11 feet 10 ins.	12 feet 4 ins.		...	20 feet	...
Draught	9 feet 2 ins.	9 feet 8 ins.		...	11 feet 10 ins.	...
Power of oil-fuel, surface, engines	400 E.H.P.	600 E.H.P.		1,200 E.H.P.	1,800 E.H.P.	2,500 E.H.P.
Power of electric under-water motors	240 E.H.P.	320 E.H.P.		600 E.H.P.	800 E.H.P.	...
Maximum speed on surface	11 knots	12 knots		15 knots	16 knots	17 knots.
Maximum speed submerged	8 knots	8.5 knots		9 knots	10 knots	...
Radius of action on surface	......	1,200 miles at 9 knots.		...	1,500 miles at 12 knots.	...
Radius of action submerged	......	50 miles at 9 knots.		...	70 miles at 6 knots.	...
Armament	One torpedo tube.	Two tubes	Two tubes	Two or three tubes.	Four tubes	...
	Three 17.7-in. torpedoes.	Four 17.7-in. torpedoes.	Four 17.7-in. torpedoes.	Four or six torpedoes.	Eight 19.6-in. torpedoes.	...
				One 1.456-in. gun.	Two 3.464-in. guns.	...

these latter, eight were completed when war was declared, the six others—*U 33* to *U 38*—commenced in 1913, were certainly not finished when hostilities opened. On the other hand, Germany added to her own number of submarines five Austrian and one Norwegian boats, which were at the time almost completed at the Krupp Germania Yard, Kiel. The German Navy, therefore, had in all 38 submarines at the opening of hostilities. The first German submarine—*U 1*—was launched at the Germania Yard on August 30, 1905; this is largely an imitation of the French boats of the *Aigrette* type, commenced in 1902. The accompanying table gives some particulars of the German boats. As regards some of the later boats, particulars were given in the "Navy Annual." Length, 214 feet 1¾ inches; beam, 20 feet; displacement on surface, 750 metric tons; submerged, 900 tons; reserve of flotation at surface displacement, 56 per cent.; speed on surface, 20 knots, submerged, 10 knots; horsepower of oil motors, 4,000 brake horsepower, working on twin screws. It is thought that the five Austrian boats which Germany has kept have the same dimensions as the *U 33* to *U 38* boats. The Norwegian boat may be considered a replica of the *U 9* type. It is stated that Germany put in hand 20 new submarines at the end of 1914. In the summer of 1907 Germany had only one submarine in the service and seven in course of construction. The sum set apart for submarine construction in the 1907 budget amounted to £250,000. It increased rapidly, amounting to £350,000 in the budget for 1908; to £500,000 in that for 1909; to £750,000 in each of those for 1910, 1911 and 1912; to £1,000,000 in that for 1913; and to £950,000 in that for 1914. Germany undertook late in the day the construction of submarines, since her first boat, the *U 1*, launched in August, 1905, only entered the service in February, 1907. By this means the German Navy was able to benefit by the experience gained elsewhere. Her first twelve boats resembled the French type *Aigrette;* the eight following boats, of a larger type, resembled the larger Pluviose class. M. Laubeuf adds that, for equal displacement and age, the French submarines have better nautical qualities and a more powerful armament than the German ones. Notwithstanding the great activity displayed by Germany in submarine construction since 1907 and the importance of the sums voted for this work, she has not had time to build a very large number of submarines. Her program was to form by 1917 a flotilla of 72 boats; by the opening of hostilities she was able to put in service slightly more than half that number. Austria commenced the construction of submarines in 1907. When war was declared, she had six ready, *U 1* to *U 6*, of 300 tons, and four, on the Krupp designs, under construction at Pola. The five built by Krupp, above referred to, were to be numbered *U 7* to *U 11*. The four Pola boats have been completed, and are numbered *U 12* to *U 16*.—"Engineering."

JAPAN.

NAVAL OUTPUT OF JAPANESE SHIPYARDS IN 1914.

The most interesting vessel built in 1914 was the battleship *Fuso*, of 30,600 tons displacement, which was launched at the building dock at Kure in March, 1914. This vessel, which is one of the largest and most powerful battleships afloat, carries twelve 14-inch guns in six turrets, all arranged on the centerline of the ship. The vessel will go into commission during the latter part of next year.

In addition to the *Fuso* there are under construction three sister ships, namely, the *Yamashiro*, at the Yokosuka Navy Yard, the *Hiuga* at the Mitsu Bishi Dockyard at Nagasaki, and the *Ise* of the Kawasaki Dockyard of Kobe. The propelling machinery of these ships consists of Brown-Curtis turbines and Japanese Admiralty type water-tube boilers, with the

exception of the *Hiuga*, which is fitted with Parsons turbines. The battle cruiser *Hiyei* was completed last summer and is now engaged in actual service, while her two sister ships, the *Kirishima* and *Haruna*, are now undergoing speed trials at the Mitsu Bishi Dockyard and the Kawasaki Dockyard, respectively.

After the outbreak of the present war, a supplementary budget was passed in the special session of the Diet, held last September, authorizing the construction of ten destroyers of 600 tons displacement and 9,500 indicated horsepower, to be completed in about six months. Contracts for the construction of two of these vessels were placed with the Mitsu Bishi Dockyard, two with the Kawasaki Dockyard, and one each with the Osaka Iron Works and the Uraga Dock Company. The hulls, machinery and equipment of these vessels will be constructed entirely in Japan.—" International Marine Engineering."

PORTUGAL.

THE PORTUGUESE TORPEDO-BOAT DESTROYER *DOURO*.

The Portuguese Government, desiring to undertake the building of torpedo-boat destroyers at its arsenal at Lisbon, arranged a contract with Yarrow & Co., Limited, to supply it with complete designs of the hull so that the vessels could be constructed in Portugal, Yarrow & Co., Limited, supplying all the machinery and boilers and guaranteeing the full-speed and other trials.

The contract was for two vessels, of which the *Douro* was the first and the *Guadiana* the second. The first-named has recently completed all her trials and is now in the Portuguese service. So satisfactory have been the results that the Government has put in hand two identical vessels, the *Vouga* and the *Tamega*, for which the British firm is supplying the entire machinery equipment.

It is worthy of note that the success of the *Douro* is in a very great measure due to the high class of workmanship performed by the Portuguese workmen at Lisbon, together with the necessary supervision by the naval authorities. The hull of the vessel will stand the closest inspection, the plating and riveting being of the very highest class throughout.

The design of the *Douro* type followed that of the ten Brazilian destroyers constructed a short time ago by Yarrow & Co., Limited, but reciprocating machinery was discarded in favor of turbines. The length of the vessel is 240 feet; beam, 23 feet 6 inches; fully loaded displacement, 670 tons; and nominal shaft horsepower, 11,000. She is propelled by three screws driven by Parsons' turbines, constructed by Yarrow & Co., Limited, the high-pressure turbine being arranged on the center shaft, and the two low-pressure turbines with the reversing turbines on the wing shafts. There are three water-tube boilers of the latest Yarrow type designed for burning coal, one being double-ended and the other two single-ended. The only fuel used is coal. The question of oil fuel was considered, but, as its supply might not have been sufficiently reliable, coal was considered preferable.

The armament consists of one 4-inch quick-firing gun and two 12-pounder guns, as well as two 18-inch torpedo tubes, one of which is arranged for firing over the stern. Accommodation is arranged for five officers, and the total complement is seventy-two.—" The Engineer."

ASSOCIATION NOTES.

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(RANK OF REAR ADMIRAL.)

Engineer-in-Chief of the U. S. Navy, 1861-1869.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXVII. AUGUST, 1915. No. 3.

HEAT LOSSES IN STEAM TRANSMISSION.

BY W. L. CATHCART, MEMBER.

The action during the cooling of saturated steam is complex. Unless the steam velocity is high, a film of condensate adheres to the inner surface of the pipe. This film and that surface are usually assumed to be at the temperature corresponding with the pressure in the pipe, owing to the heat radiated from the column of steam. The film, however, acts also as an insulator to heat conduction from the steam, since the conductivity of water is, according to Lord Rayleigh, but $\frac{1}{115}$ that of iron. Further, between film and pipe, pipe and covering, and the covering and the surrounding air, external conduction takes place, meeting resistance at these three pairs of contact surfaces.

These various increments of resistance to heat transmission are shown roughly in Fig. 1. Let *Aa* be this resistance at the middle of an uncovered steam column, from which heat is dissipated chiefly by radiation but partially by conduction. Then, *Bb*, will be the resistance at the interface with the film. The low conductivity of the film increases the resistance to *Cc*, external conduction to the pipe adds the increment *cd*, internal conduction through the pipe metal makes the aggregate resistance *Ee*, external conduction to the covering

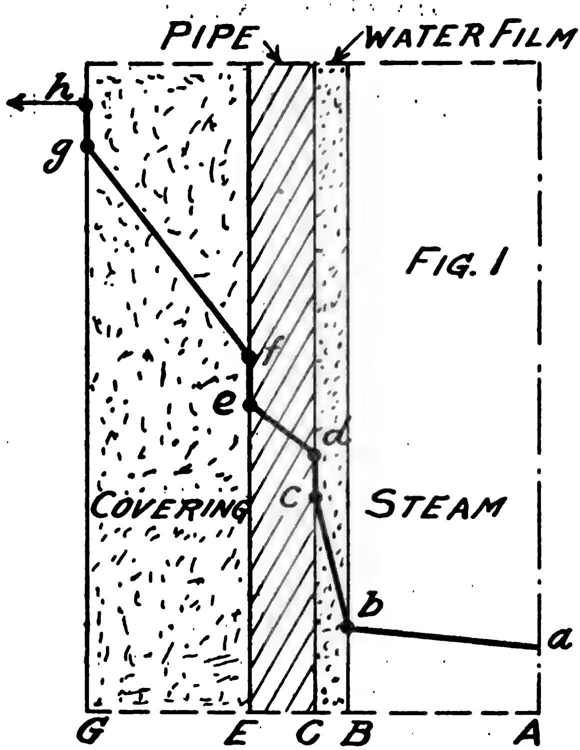


FIG. 1

increases the resistance by the amount *ef*, the passage of the covering raises the total to *Gg*, and external conduction to the surrounding air gives the final increment *gh*. The total resistance to heat transference from the middle of the column is then represented by *Gh*.

I. RADIATION AND CONDUCTION.

Heat is transmitted from one body to another by conduction, radiation, or both. The fundamental difference between these two processes is that the former proceeds by the direct agency of matter—the contact or other action of one molecule upon another—while, in radiation, there is no connecting medium perceptible to the senses between a hot radiating body and the cooler ones which absorb its heat.

The Stefan-Boltzmann law of radiation is: the radiation

of a black body is proportional to the fourth power of its absolute temperature, *i. e.*, if T be the absolute temperature of such a body, and T_0 that of another body to which radiation takes place, then, since radiation is a mutual action, the total energy radiated by the former body will be:

$$R = K(T^4 - T^4_0),$$

in which R, T, and T_0 may be expressed in any system of units and the value of the constant K depends on that system. This law was based on experiments by Stefan and later proved theoretically by Boltzmann. It has received ample experimental support. The difficulty in its practical use lies in giving K a proper value and in finding the temperature of the surface of the radiating body.

The principal factors governing radiated energy are:

(*a.*) *The Temperature Difference.*—As Stefan's law indicates, the higher relatively the absolute temperature of a radiating body, the greater proportionately will be the amount of energy radiated. With increasing temperature, the radiating power of all bodies tends to follow Stefan's law more closely.

(*b.*) *Condition of Radiating Surface.* — For surfaces of various characters, this condition is met by suitable values of the constants in the formulas. C. L. Norton* gives a practical example of the effect of pipe-surface on heat-loss during steam transmission. Stating the total heat-loss in B.t.u. per square foot of pipe-surface per minute, he found:

Condition of pipe-surface.	Heat-Loss.
New pipe,	11.96
Fair condition,	13.84
Rusty and black,	14.20
Cleaned with caustic potash, inside and out,	13.85
Painted dull white,	14.30
glossy white,	12.02
Cleaned with potash again,	13.84
Coated with cylinder oil,	13.90
Painted dull black,	14.40
glossy black,	12.10

* "Trans. Am. Soc. Mech. Engs.," Vol. XIX.

As conduction does not depend on the nature of the surface, the differences between these various total losses are due solely to changes in the amount radiated owing to the altered surface of the pipe. The temperature was that corresponding to 200 pounds pressure.

Internal Conduction.—Consider one surface of a metal plate of uniform thickness as maintained at the constant temperature t_1 and the other at the constant and lower temperature t_2. Let x be the thickness and A the area. Then, in the time T, the quantity of heat transmitted through the plate will be:

$$H = k\frac{(t_1 - t_2)\,AT}{x} \text{ and } k = \frac{Hx}{AT(t_1 - t_2)} \quad . \quad . \quad (1)$$

in which the constant k is the mean *internal conductivity* of the metal between the given temperatures. The reciprocal $\frac{1}{k} = r$ is the *internal thermal resistance.* These constants are coefficients whose absolute value depends on the system of units from which they were derived. They also vary with temperature.

The absolute conductivity of wrought iron, as deduced from the experiments of Forbes*, is 480.78 B.t.u. per hour, per square foot of plate one inch thick, per degree F. temperature-difference between the receiving and emitting surfaces.

Jaeger and Diesselhorst†, for the same system of units, obtained for wrought iron conductivities of 430.18 at 64.4 degrees F., and 406.77 at 212 degrees F. These values show, like the experiments of Forbes, the decrease in the conductivity of iron with increase in temperature.

Let $t_1 - t_2 = t$. Then:

$$t = \frac{Hx}{ATk} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

which is practically an inconsiderable amount. Thus, in tests of a bare wrought-iron pipe, 0.2-inch thick with sat-

* Preston: Theory of Heat, London, 1904, p. 644.
† Situngsber. d. Berlin Akad., 1899, II, p. 746.

urated steam, it was found that 687 B.t.u. were lost per square foot per hour at a steam pressure of 81 pounds, gage. Taking k as 480, $t = 0.29$ degrees F. Again, the absolute internal thermal resistance of this pipe is $\frac{0.2}{480} = 0.00042$, which is negligible in practice.

Stefan found the conductivity of air to be 20,000 times less than that of copper, the latter being about 6.2 times that of iron. Superheated steam gives evidence of low internal conductivity. Thus, it has been found to exist unchanged in the same space, for a considerable period, with saturated steam and the water of condensation. Again, for efficient superheating, the tubes should be small, or, if large, should be fitted either with cores to form a thin annulus of steam or with deflecting plates to rotate the steam and throw the heavier and colder particles against the hot tube.

The conductivity of a pipe-covering may be materially reduced by giving it a cellular structure, providing the air-spaces are sufficiently small to minimize convection. In these conditions, air-spaces act in two ways: they increase the conductivity by giving opportunity for heat transmission by radiation, and decrease it through the low conductivity of the contained air, the net effect of the two actions being a reduction. Asbestos is *per se* a relatively good conductor; its cellular structure is what gives it efficiency as a covering.

External Conduction.—There is resistance to the flow of heat to or from any substance at its interface or joint with the emitting or receiving substance. This condition applies also to two parts of the same substance. If a rod be made of the same metal throughout, but in sections united by shrinkage fits or screw threads, there will be thermal resistance at every joint, and a solid rod of the same metal and dimensions will permit a freer flow of heat. A fusible plug melts at a lower temperature than that of the plate in which it is driven, owing to the interface resistance of the two metals.

As has been shown previously, the internal thermal resistance of a steam pipe is negligible, since the metal is capable

of conducting more heat than it can absorb or emit. The reverse is the case with pipe coverings. Their primary characteristic is low internal conductivity, and hence they can receive and deliver more heat than it is possible for them to conduct.

External conduction to or from a solid or fluid is a combination of two actions: emission by the heat-deliverer and absorption by the heat-receiver. The process is too complex for analysis. The general conditions governing it are: the temperature of the heat-delivering substance, the temperature-difference between heat-deliverer and heat-receiver, the state of rest or motion of both, the relative direction of these motions (contra-flow principle), and the character of the surfaces of the solid. In any case, the external conductivity at an interface is the reciprocal of the external thermal resistance there, and *vice versa.*

The formulae given below follow the general method of Mollier, the transfer of heat from one fluid to another through an intervening solid being divided into three stages: the external conduction from the first fluid (steam) to the solid (pipe metal), internal conduction through the latter, and external conduction from the pipe to the second fluid (air).

Let:

A = area of plate through which heat passes.
x = thickness of plate.
T = time during which heat is transmitted.
H = quantity transmitted during time T.
t_1 = temperature of steam.
t_2 = temperature of heat-receiving surface of plate.
t_3 = temperature of heat-delivering surface of plate.
t_4 = temperature of air.
k = coefficient of internal conductivity of plate.
k_1 = coefficient of external conductivity steam to plate.
k_2 = coefficient of external conductivity plate of air.
k_3 = coefficient of heat transmission, steam to air.

Since, during each of the three stages, the amount of heat transmitted is the same:

$$H = k_1 (t_1 - t_2) AT \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$H = \frac{k}{x}(t_2 - t_3) AT \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$H = k_2 (t_3 - t_4) AT \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Combining and eliminating:

$$H = \frac{1}{\frac{1}{k_1} + \frac{x}{k} + \frac{1}{k_2}} \cdot (t_1 - t_4) AT \quad . \quad . \quad . \quad (6)$$

$$H = \frac{1}{r_1 + xr + r_2} \cdot (t_1 - t_4) AT \quad . \quad . \quad . \quad (7)$$

in which r represents thermal resistance. The fraction in (7) is the reciprocal of the sum of all the resistances met by the heat in passing from steam to air, and is therefore the coefficient of heat transmission, or

$$K_3 = \frac{1}{\frac{1}{k_1} + \frac{x}{k} + \frac{1}{k_2}} = \frac{1}{r_1 + xr + r_2}. \quad . \quad . \quad . \quad (8)$$

The coefficients of external conductivity can be computed by these formulae, if the temperature of the outer surface of the pipe is known. It is impossible by ordinary methods to ascertain this temperature with scientific accuracy. As an example, take a test by Eberle * of superheated steam. The pipe was of wrought iron, 0.2 inch thick, uncovered, and the temperature of its outer surface was measured by thermo-elements. The data are:

Steam pressure, pounds per square inch, absolute	93.9
Steam temperature, degrees F	$555.8 = t_1$
Temperature of outer surface of pipe	$492.8 = t_3$
Temperature of air	$51.8 = t_4$
B.t.u. lost per square foot of outer surface per hour	$1,911.4 = H$
Steam velocity, feet per minute	5,905.8

*"Zeits. d. Ver. deuts. Ing.," April 16, 1908.

Taking $t_2 = t_3$ and A and $T =$ unity, we have from (3) and (5), $k_1 = 30.3$ and $k_2 = 4.33$.

There is a marked difference between the values of k_1 for superheated and saturated steams, as deduced from these formulae. Saturated steam cannot lose heat without partial condensation, and hence there is not only a film of condensate adhering to the interior surface of the pipe, as shown in Fig. 1, but probably the column of moving steam has also a thin casing of condensate. Formulae (3) to (8) inclusive do not consider these insulating films, since the heat transfer through them is too complex a process. With highly superheated steam, these conditions do not prevail, since the pipe, if properly protected, is hotter than the corresponding temperature of saturated steam.

Rapid relative movement of the heat-delivering and heat-receiving surfaces increases the rate of heat-transfer, because this movement keeps the temperature-difference at its maximum. This is the principle which has been applied so successfully in the water-tube boiler and whose effects are marked also in the transmission of superheated steam. With saturated steam, however, despite the scrubbing action of moving steam, the films of condensate seem to linger at moderate velocities and to nullify the tendency toward increased heat-transfer from greater velocity. For example, Eberle's series of tests of superheated steam, cited previously, give the following average data:

Number of tests	9	7
Steam pressure, pounds per square inch, absolute	95 86	95.72
temperature, degrees F.	456.0	578.7
velocity, feet per minute	1,870.7	5,905.8
Air temperature, degrees F.	51.8	51.8
Heat-loss per hour per square foot of outer surface of pipe, B.t.u.	1,370.6	2,042.2

i.e., at the same pressure, although at a higher temperature, and at a steam velocity 3.16 times greater, there was an increase in the heat-loss per square foot of 49 per cent.

On the other hand, Gutermuth* gives a table comparing

* "Zeits. d. Ver. deuts. Ing.," 1887, p. 699.

the amount of condensate of saturated steam, when the latter was stationary and when it was flowing at various velocities. The following data are taken from this table.

Duration of test, hours.	Steam press., lbs., abs.	Steam velocity, ft. per min.	Condensate per hour (lb.) with steam.		Reduction in condensate due to steam velocity, per cent.
			Flowing.	Stationary.	
8.0	71.1	1,791	398.2	398.2	0.0
4.5	69.4	3,090	352.0	374.0	5.9
6.0	53.3	4,054	202.4	215.6	6.1
9.0	71.8	9,505	129.8	233.2	44.3

All of the 17 tests, except the first at low velocity as given above, showed reduced condensate with increased velocity. These results have been confirmed by recent tests of other investigators. It is, of course, possible that, with growing velocity, more of the condensate is entrained by the steam.

II. PÉCLET'S FORMULAE FOR HEAT LOSSES.

Péclet, who modified and extended the investigation of Dulong and Petit, established formulae for the radiation and conduction-losses of a cooling body which represent, in fair approximation, the actual results, and which, with a suitable correction, can therefore be used in practice.

The need of this correction arises from the fact that the investigations of these physicists, although most elaborate, were conducted within restricted limits of temperature and with special apparatus of small size, so constructed that all conditions could be regulated with certainty. Further, the cooling bodies employed by Péclet were metallic vessels filled with hot water, the coefficient of emissivity of which differs somewhat from that of steam. On the other hand, Russner found that results obtained by the use of the radiation factor determined by Péclet compared favorably with those from Stefan's law, Stefan himself showed that there was a reasonable agreement between his law and Dulong and Petit's formula

on which that of Péclet is based, and Pasquay, a pioneer in the invention and use of pipe-coverings, regards Péclet's formulae as satisfactory bases for computing the minimum loss in any given case. With regard to this, he says:*

"The values which we obtain from Péclet's formulae for the heat-losses of bare pipes are less than those found in practice, but these formulae were based on rigidly exact laboratory tests and assume still air, a condition which does not exist ordinarily even in enclosed spaces, still less in power plants or in mill rooms where there is movement of men and machinery or in passages or tunnels where the temperature-difference always causes a certain motion of the air, the intensity of which cannot be determined accurately. * * * It is important, however, to have formulae which (like Péclet's), will give us the minimum loss under the most favorable conditions."

Péclet limited his tests to those on bodies cooling in air at atmospheric pressure in an air-tight chamber with dull walls. As cooling bodies, he employed rectangular and spherical vessels of thin brass, filled with hot water kept continuously agitated. The rate of cooling, at a constant temperature of the air in the chamber, was found for the vessel when bare and when covered with various substances. The temperature-difference between cooling body and air, ranged from 45 degrees to 117 degrees F., with an average air temperature of 53.6 degrees. Within these limits, his results were in entire accord with the formulae of Dulong and Petit, and hence he decided that their formulae were exact up to the limit of temperature-difference, 468 degrees F., used by them. The important results of his experiments were the determination of the value of the coefficients for use in practice. As the formulae are empirical, the range fixed by Péclet should not be exceeded.

Heat Loss by Radiation.—Péclet's conclusions as to radiation were:

"The quantity of heat, per unit of surface and of time,

* "Warmeschutz im Dampfbetrieb:" Wasselnheim, 1900, p. 7.

emitted by radiation is independent of the size and form of the body—if the surface has no reëntrant portions (since radiation is a mutual action). This quantity depends only on the nature of the surface, on the excess of its temperature above that of the surrounding objects, and on the absolute value of the temperature of the latter."

His formula* is:

$$R = 124.72\, K\, a^{\theta}\, (a^{t} - 1) \quad . \quad . \quad . \quad . \quad . \quad (9)$$

in which, R is the quantity of heat in calories (large) emitted by radiation per square meter of surface per hour, t is the temperature-difference between this surface and the surroundings, θ is the temperature of the latter, *i.e.*, usually that of the air, a is a constant having for metric units the value 1.0077, and K, the radiation factor, is a number whose magnitude depends on the nature of the radiating surface. The temperatures are in degrees C.

This formula is too complex for ready computation, and hence Péclet tabulated the numerical values of the products of all the factors except K in the right-hand member of (9), for an air temperature of 15 degrees C. and temperature differences between 10 degrees and 250 degrees. The coefficients of K thus found have been reduced to British units and are given in Table 1. Below each coefficient is set the amount per degree F. of temperature difference to be added to that coefficient through the range between it and the next; at the foot are placed the factors by which the tabulated coefficients are to be multiplied for other coefficients.

The values of K, as determined by Péclet, are given in Table 2. The figures for paper and cloth apply to all colors.

From Table 1 and formula (9), it will be seen that the radiating power of a body increases slowly with the temperature of the surroundings and rapidly with the temperature-difference. Table 2 shows the marked advantage for preventing radiation, of a polished casing, especially as the greater part of the heat loss is due to this cause.

* "Traité de la Chaleur," Paris, 1860, I., p. 373.

TABLE 1.—HEAT LOSS BY RADIATION.

(Values of R in B.t.u. per square foot per hour of radiating surface per hour at an air temperature of 59 degrees F.)

Excess (t^o) of temperature of radiating surface above air temperature.	$R = K \times$	Excess (t^o) of temperature of radiating surface above air temperature.	$R = K \times$
Degrees F.		*Degrees F.*	
18	20.16	252	451.1
	1.2		3.26
36	41.76	270	543.78
	1.289		3.694
54	64.96	288	610.27
	1.403		3.836
72	90.22	306	679.32
	1.518		4.11
90	117.54	324	753.3
	1.642		4.472
108	147.1	342	833.8
	1.761		4.798
126	178.79	360	920.16
	1.915		5.181
144	213.26	378	1,013.42
	2.021		5.607
162	249.64	396	1,114.34
	2.261		6.034
180	290.34	414	1,222.96
	2.405		6.53
198	333.63	432	1,340.5
	2.597		10.403
216	380.38	450	1,527.75
	2.801		
234	430.79		
	3.017		

For air-temperature of	32°	50°	68°	86°	104°	122°	140°	158°	176°	194°	212°
Multiply above values by	.89	.96	1.04	1.12	1.21	1.31	1.41	1.52	1.65	1.78	1.92

TABLE 2.—VALUES OF RADIATION FACTOR K.

(B.t.u. per square foot per hour per 1 degree F. of temperature difference.)

Silver, polished	.027	Powdered charcoal	.701
Copper	.033	Plaster	.738
Tin	.044	Wood	.738
Zinc, polished	.049	Cotton cloth (canvas)	.749
Brass	.053	Woolen cloth	.754
Sheet lead	.133	Silk	.761
Sheet iron, polished	.092	Oil paint	.761
Sheet iron, ordinary	.568	Paper	.773
Sheet iron, rusted	.689	Lamp-black	.822
Cast iron, new	.65	Water	1.089
Cast iron, rusted	.689	Oil	1.484

Heat Loss by Conduction.—Péclet's conclusions as to heat transmission by conduction to the air are:

"The loss of heat which is due to the contact of the air is independent of the nature of the surface of the body and of the temperature of the surrounding air. It depends only on the excess of the temperature of the body above that of the air and upon the form and dimensions of the body."

His formula for this loss is:

$$A = 0.552\, K'\, t^{1.233} \quad . \quad . \quad . \quad . \quad . \quad (10)$$

in which, A is the quantity of heat in calories (large) thus transmitted per square meter of surface per hour, t is the temperature difference between the surface and the surround-

TABLE 3.—HEAT LOSS BY CONDUCTION TO THE AIR.

Values of A in B.t.u. per sq. ft. of conducting surface per hour.

Excess ($t°$) of temperature of the surface above air-temperature.	$A = K' \times$	Excess ($t°$) of temperature of the surface above air-temperature.	$A = K' \times$
Degrees F.		*Degrees F.*	
18	16.92	252	439.99
	1.28		2.166
36	39.96	270	478.98
	1.44		2.206
54	65.88	288	518.69
	1.56		2.226
72	93.96	306	558.76
	1.64		2.276
90	123.48	324	599.72
	1.738		2.288
108	154.76	342	640.91
	1.804		2.334
126	187.24	360	682.92
	1.854		2.359
144	220.61	378	725.38
	2.09		2.381
162	258.23	396	768.24
	1.804		2.4
180	290.7	414	811.44
	2.		2.416
198	326.7	432	854.93
	2.058		2.354
216	363.74	450	897.3
	2.1		
234	401.54		
	2.136		

ing air, and K' is a number whose magnitude varies with the form and dimensions of the body.

Péclet computed from (10) the products of all factors except K' in the right-hand member, for temperature-differences as in Table 1. The coefficients thus found, reduced to British units, are given in Table 3 with also the differences per degree F.

The value of the conduction-factor K' depends on both the form and dimensions of the cooling body, owing to differences in the direction and intensity of convection currents. Péclet's formulae are:

For horizontal cylinders,

$$K' = 0.421 + \frac{0.308}{r} \quad . \quad . \quad . \quad . \quad . \quad (11)$$

For vertical cylinders,

$$K' = \left(0.149 + \frac{0.0443}{\sqrt{r}}\right)\left(2.43 + \frac{1.585}{\sqrt{h}}\right) \quad . \quad . \quad (12)$$

For cylinders inclined at the angle α to the horizontal,

$$K' = K' \text{ (hor.) } \cos \alpha + K' \text{ (vert.) } \sin \alpha \quad . \quad (13)$$

In these formulas, r is the radius in inches and h the height in feet. Tables 4 and 5 give values of K' computed from formulae (3) and (4).

TABLE 4.—CONDUCTION FACTOR K' FOR HORIZONTAL CYLINDERS.

Diameter, inches.	K'.	Diameter, inches.	K'.
2	0.729	12	0.472
4	0.575	14	0.465
6	0.524	16	0.459
8	0.498	18	0.455
10	0.483	20	0.452

The formulae and tables show that the conduction-loss is considerably smaller than the corresponding radiation-loss and that the former grows much more slowly than the latter with increased temperature-difference.

TABLE 5.—CONDUCTION FACTOR K' FOR VERTICAL CYLINDERS.

Diameter, inches	Height of cylinder, feet.		
	5	10	20
2	0.607	0.567	0.537
4	0.565	0.528	0.501
6	0.549	0.513	0.487
8	0.537	0.501	0.476
10	0.530	0.495	0.470
12	0.524	0.490	0.465
18	0.515	0.481	0.457
24	0.509	0.475	0.451

Total Heat Loss.—The total loss is the sum of the values given by (9) and (10), or

$$M = R + A \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Example: Uncovered wrought-iron pipe 6.3 inches outside diameter, saturated steam, pressure 186 pounds, absolute-temperature of steam, 375.4 degrees, of air, 87.1 degrees, temperature-difference, 288.3 degrees. Value of k (Table 2) = 0.568.

Radiation Loss, R.—(Tables 1 and 2).

At 59° air-temp. and 288° temp.-diff., . $R = 680.27\,K$
Addition for 0.3° = 3.836 × .3 = . . 1.15

At 59° air-temp. and 288.3° temp.-diff. . $R = 611.42\,K$

Factor for air-temp. 87.1° = 1.12 + 0.005 (87.1 — 86) = 1.125.
∴ at 87.1° air-temp., $R = 611.42 \times 1.125 \times 0.568 = 390.7$ B.t.u.

Conduction Loss, A.—From Table 3, at 288° temp.-diff., $A = 518.69\,K'$; the difference per degree is 2.226. From Table 4, for a 6.3-inch pipe:

$K' = 0.524 - 0.3\ (0.524 - 9.509) = 0.52.$

At 288° temp.-diff.	$A = 518.69\ K'$
Addition for $0.3° = 0.3 \times 2.226 =$ . .	0.67
At 288.3° temp.-diff.	$A = 519.36\ K'$

$\therefore A = 519.36 \times 0.52 = 270.07$ B.t.u.

Total Heat Loss: $M = R + A = 390.7 + 270.07 = 660.77$ B.t.u. per square foot per hour.

This is the minimum heat loss computed for the conditions of Péclet's experiments. As shown by Table 6, the correction, for the conditions of practice, ranges between 0.1 and 0.47. Taking 0.35 in this case, we have

Estimated Heat Loss $= 660.77 \times 1.35 = 892.04$ B.t.u. per square foot per hour, which is 96 per cent. of the actual heat loss as found by measuring the condensate.

III. Heat Losses of Uncovered Pipes.

Saturated Steam.—In Table 6, there are assembled the data of 19 of the principal tests of uncovered pipes containing saturated steam. The steam was in all cases motionless, except for the flow necessary to replace the condensate. The heat loss was considered to be the latent heat of the amount of this condensate which was properly chargeable to the pipe under test. The condensate was assumed to be discharged at the temperature of the steam. The loss per square foot of outer surface of pipe, as given in column 8 of the table, refers in general to that surface only, and not to the flanges and valve chambers of a steam pipe in actual service.

Column 9 gives the total losses as computed by Péclet's formulae, and column 10 shows that the average correction to be applied to the latter is 0.3. The necessity for this correction is obvious for two reasons: first, Péclet's results were derived from laboratory experiments on apparatus of limited size, under wholly controllable conditions as to air-currents and the exact measurement of the heat dissipated; second,

TABLE 6.—HEAT LOSSES OF SATURATED STEAM IN UNCOVERED PIPES.

Test No.	Pipe.		Steam pressure, lbs., abs.	Temperature, F.		Temp-diff. steam and air, deg. F.	Heat-loss per sq. ft. per hour, B.t.u.			
	Metal.	Diam., outside, ins.		Steam.	Air.		Actual.	Péclet.	Ratio $\frac{A}{P}$.	Per 1° F. of temp.-diff.
1	C.I.	6.2	74.7	307.1	76.1	231.0	596.02	502.44	1.19	2.58
2	"	2.39	74.2	306.6	75.4	231.2	642.74	566.38	1.14	2.78
3	"	2.39	94.7	323.7	50.5	273.2	841.46	681.05	1.23	3.08
4	"	2.39	164.7	365.7	60.5	305.2	1019.37	827.16	1.23	3.34
5	W.I.	3.53	17.7	221.5	64.8	156.7	360.63	287.82	1.25	2.301
6	"	3.53	39.7	266.7	69.2	197.5	433.9	393.35	1.10	2.197
7	"	6.3	47.4	277.3	78.3	199.0	528.41	379.13	1.39	2.655
8	"	3.0	45.1	274.4	61.0	213.4	574.16	438.69	1.31	2.691
9	"	3.53	54.7	286.5	68.5	218.0	595.7	448.41	1.33	2.732
10	"	6.3	96.0	324.7	89.1	235.6	687.45	492.65	1.39	2.918
11	"	3.53	73.7	306.2	62.6	243.6	697.19	519.21	1.34	2.862
12	"	3.53	84.7	315.8	69.0	246.8	785.4	534.24	1.47	3.182
13	"	3.0	93.7	322.9	63.0	259.9	753.87	582.13	1.29	2.901
14	"	3.53	100.7	328.1	68.0	260.1	775.9	576.24	1.35	2.983
15	"	8.5	125.2	344.3	75.5	268.8	728.45	563.49	1.29	2.71
16	"	3.53	114.7	337.7	65.3	272.4	859.6	613.39	1.40	3.156
17	"	7.64	164.0	365.4	92.3	273.1	807.4	606.55	1.33	2.956
18	"	6.3	185.9	375.4	87.1	288.3	933.2	660.77	1.41	3.237
19	"	3.0	185.5	375.3	71.1	304.2	962.35	748.31	1.29	3.163
				Means =		246.22			1.302	2.864

the possible errors of the condensate method under practical conditions, if not applied with extreme accuracy.

With regard to tests in general, the possible reasons affecting this lack of agreement are: (*a.*) Air currents not due to convection. This is a disturbing influence which exists to a greater or less extent in all tests under ordinary conditions. (*b.*) Inaccurate measurement of the amount of the condensate which is properly chargeable to heat loss from the pipe. Condensate produced by the attachments of the test-pipe should be carefully computed. Defective action of the separator will affect the amount of the condensate. Leaky steam traps will allow steam to filter through. Uncovered drip pipes and traps add unduly to the amount of the condensate. Unsuitable test apparatus. Some investigators seem to have disregarded the fact that radiation is a mutual action, and, further, that convection currents from two adjacent heated bodies may conflict. (*d.*) Minor causes are: inaccurate measurement of the loss due to flanges, valve chambers, etc., and relying on surface measurements—which are wholly untrustworthy—for the temperatures of the outer surface of a pipe or covering. The temperature of any such surface is not that of the film of air adjacent to it, which is what a thermometer measures to a greater or less extent.

TABLE 7.—HEAT LOSS IN B.T.U. PER SQUARE FOOT OF OUTER SURFACE OF PIPE PER HOUR.

Test No.	By radiation.			By conduction.			Total.		
	Stefan (Eberle).	Péclet.	Ratio $\frac{S}{P}$	Eberle.	Péclet.	Ratio $\frac{E}{P}$	Eberle.	Péclet.	Ratio $\frac{E}{P}$
8	305.02	214.34	1.42	269.14	224.35	1.12	574.16	438.69	1.31
13	424.04	295.97	1.43	329.83	286.15	1.15	753.87	582.12	1.29
19	571.35	401.04	1.42	391.	347.27	1.13	962.35	748.31	1.29
7	294.15	208.17	1.41	234.26	170.96	1.37	528.41	379.13	1.39
10	404.88	282.07	1.44	282.57	210.58	1.34	687.45	492.65	1.39
18	555.46	390.7	1.42	377.74	270.07	1.39	933.2	660.77	1.41
	Means:		1.42			1.26			1.35

That there is a definite agreement between Péclet's results,—when suitably corrected for practical conditions,—and Stefan's law is shown by the data of Table 7.

These tests* are part of an extensive series conducted by Eberle under the auspices of the "Vereines deutscher Ingenieure." The total heat loss was measured by the condensate method. The radiation loss was computed by Eberle from Stefan's law, in the form:

$$R = 0.1406\left[\left(\frac{T}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right]$$

in which, R, the radiation loss, is given in B.t.u. per square foot of pipe surface per hour and T and T_0 are respectively the absolute temperatures F. of the pipe and the air. The value of the constant K in Stefan's law was thus assumed arbitrarily to be 0.1406×10^{-8}. The conduction loss was taken as the difference between the total loss and that computed for radiation.

Although the steam pressure (Table 6) differed in each of the tests of Table 7, column 4 shows that the calculations by Stefan's law are almost uniformly 42 per cent. higher than those by Péclet's formulae, *i. e.*, that there is a constant relation between the two. The value of Eberle's constant may have been too high, that of Péclet's radiation constant too low, or both may be right and the difference may be due to that between the conditions of the two series of tests. The same uniformity does not exist in the figures for the total and conduction losses, since both depend on condensate measurements, with the errors possible in that method.

From Table 6, the total heat loss can be calculated, in fair approximation for any given case within the temperature range of that table, by computing it from Péclet's formulae and multiplying the result by the mean ratio given at the foot of column 5; or, by assuming the outside of the pipe to be at the temperature of the steam, and multiplying the difference

*"Zeits. d. Ver. deuts. Ing.," April 4, 1908.

between this temperature and that of the air by the factor given in column 6.

Superheated Steam.—Tests to find the heat loss of superheated steam in transmission are more difficult than those with saturated steam, because the steam velocity is a factor. For practical results, tests should be made only on moving steam at the velocity at which it is desired to ascertain the heat loss. The various factors of this complex problem are:

(*a.*) *Absolute and Proportionate Heat Losses.*—The absolute heat loss is the loss per square foot of outer surface of pipe per hour. The proportionate loss is the loss of heat per pound of steam transmitted. With higher velocity, the absolute loss is divided among a greater number of pounds of steam, making the proportionate loss less.

(*b.*) *High and Low Degrees of Superheat.*—According to the investigation of Richter*, highly superheated steam simply loses some of its superheat during transmission. Hence, the only method of ascertaining its heat losses is to find the average quality of the steam at entrance to and at exit from the pipe. This determination is difficult, since it involves temperature measurements at various points in the cross-section of the column of steam. With low superheat an absolutely accurate determination is impossible, owing to the formation of condensate, some of which will be entrained by the swiftly moving steam, and, passing through the separator, will escape measurement.

(*c.*) *Temperature of the Pipe.*—Pipe temperature is the principal factor affecting heat loss. With saturated steam, this temperature is very close to that of the steam. With superheated steam, there is a marked difference between the two temperatures, the temperature fall from the center to the circumference of the pipe depending mainly on the steam velocity. At high velocity, the steam has little time to cool and the proportionate loss is less than at low speed. But the temperature fall is also less, the pipe temperature higher, and

*"Zeits. d. Ver. deuts. Ing.," Feb. 24, 1906.

the absolute loss greater. At low velocity these effects are reversed throughout.

In view of these various factors, it is clear that with superheated steams at the same temperature there may be absolute and proportionate losses which will differ widely. Tests based on steam temperature or steam velocity only are of no service. The number of available tests is limited. The data* given in Table 8 are fairly representative for the velocity and temperature range tabulated. These tests were made on an uncovered pipe, 2.76 inches internal diameter and 87 feet 2 inches long. The pressure during the series averaged 96 pounds, absolute. The coefficient k_1 of external conductivity of steam to pipe metal was taken as 30.75.

TABLE 8.—HEAT LOSSES OF SUPERHEATED STEAM IN BARE PIPES.

Computed for a steam velocity of 4,920 feet per minute and an air temperature of 68 degrees F.

Temperature, degrees F. of		Temperature-difference between steam and		Heat loss per square foot of outer surface of pipe per hour, B.t.u.
Steam.	Pipe.	Air.	Pipe.	
347	320.0	279	27.0	823
392	359.6	324	32.4	1,018
437	397.4	369	39.6	1,225
482	435.2	414	46.8	1,450
527	471.2	459	55.8	1,697
572	509.0	504	63.0	1,956
617	545.0	549	72.0	2,232
662	579.2	594	82.8	2,531
707	615.2	639	91.8	2,841
752	649.4	684	102.6	3,170

It should be observed that the temperature of the pipe was found, for the assumed value of k_1, by equations (3), etc., as given herein, and the heat losses by applying the same methods as those used for saturated steam, thus assuming that for the same pipe temperature the heat-emissivities of stationary saturated steam and of superheated steam at the

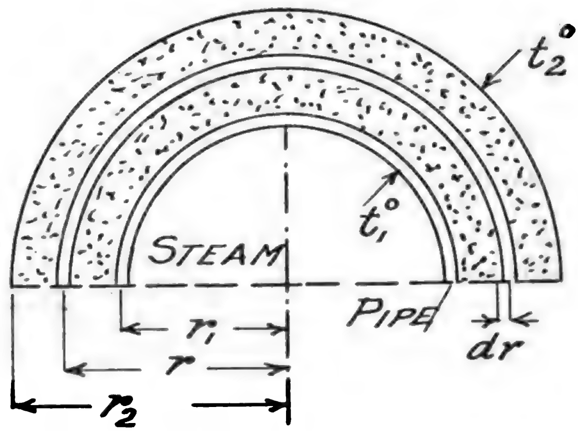
* "Zeits. d. Ver. deuts. Ing." March 28, 1908.

velocity given, are the same. These assumptions affect the absolute accuracy of the results, but their effects are minor.

IV. HEAT LOSSES OF COVERED PIPES.

By the use of Péclet's formulae, the efficiency of a covering can be ascertained, if the conductivity is known, and the conductivity, if the heat loss has been found, both in reasonable approximation. The general method is as follows:

FIG. 2



In Fig. 2, let r_1 and r_2 be respectively the internal and external radii of the covering, t_1 and t_2 the corresponding temperatures, θ the temperature of the air, C the conductivity, and M the total heat loss. This loss must be equal to the amount of heat traversing an elementary annulus of radius r, width dr, and temperature fall dt. Then, by equation (1), we may write:

$$M = -\frac{2\pi r\, Cdt}{dr}, \text{ from which}$$

$$Cdt = -\frac{Mdr}{2\pi r}. \text{ Integrating between the lim-}$$

its, t_1, t_2, and r_1, r_2, and transforming:

$$M = \frac{2\pi C(t_1 - t_2)}{N} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

in which, $N = m(\log r_2 - \log r_1)$ and $m = 2.3025$, the modulus of the Napierian system.

Again, Newton's law of cooling is true for very small temperature-differences only. Assuming that it is universally true, we may write:

$$M = 2\pi r_2 Q(t_2 - \theta) \quad . \quad . \quad . \quad . \quad . \quad (16)$$

in which $Q = K + K'$, *i. e.*, the sum of the radiation and conduction coefficients for the covering, as given in Tables 2, 4 and 5. Combining (15) and (16), we have:

$$M = \frac{2\pi r_2 QC(t_1 - \theta)}{C + Qr_2 N} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

which is the general formula giving the approximate heat loss on the assumption as above with regard to Newton's law.

Analysis of this formula will show that the heat loss does not decrease inversely as the thickness of the covering but in a less degree, since the extent of surface exposed to radiation and conduction is a factor. Hence, doubling the thickness will reduce the loss by nearer one-third than one-half of the original amount. Again, it can be shown that, when the value of C is very low, M decreases rapidly with growing thickness, until a limit is reached when the increased size of the radiating surface will neutralize the resistance of low conductivity and M will grow with greater thickness.

Example.—Horizontal cast iron pipe, 3 inches nominal diameter; saturated steam at 293 degrees F.; air temperature 59 degrees; covering, one inch thick of asbestos cased with canvas; conductivity C of covering = 0.48 for block one foot square and one foot thick.

Heat Loss of Bare Pipe.—Temperature of pipe = that of steam; temperature-difference = 293 − 59 = 234 degrees; for cast iron (Table 2) $K = 0.65$. Radiation loss (Table 1) = 430.79 × 0.65 = 280.01. By formula (11), $K' = 0.626$. Conduction loss (Table 3) = 401.54 × 0.626 = 251.36. Total heat loss = 280.01 + 251.36 = 531.37 B.t.u. per square

foot of pipe surface, and, per running foot of pipe = 416.6 B.t.u.

Heat Loss of Covered Pipe.—$t_1 - \theta = 293 - 59 = 234$ degrees. Taking all dimensions in feet and pipe as nominal size, $r_2 = 0.229$ foot, $C = \frac{0.48}{12} = 0.04$, $N = 0.451$. $Q = K + K'$, these constants being for the covering. By Table 2, K for canvas $= 0.75$. By formula (11), $K' = 0.544$. Hence, $Q = 1.294$. Substituting in formula (17):

M (per square foot of covering surface) $= 97.7$ B.t.u.

M (per running foot of covering surface) $= 127$ B.t.u.

$$\text{Economy of covering} = \frac{(416.6 - 127)}{416.6} = 70 \text{ per cent.}$$

These values of M are approximate, because we have assumed Newton's law to be true for all temperature-differences, and, on this assumption, have taken the coefficients of K and K' as unity and $Q = K + K'$. This assumption is valid only for very small differences of temperature between the cooling body and the surrounding air.

The true value of M can be found, however, from these results by a series of approximations, since from the approximate M just deduced, we can ascertain the corresponding approximate temperature of the surface of the covering. The method is as follows:

Data.—First value of $M = 127$; area of one running foot of surface of covering $= 1.44$ square feet; radius of outside of covering $= 0.229$ foot; temperature $t_1 = 293$ degrees; air temperature $= 59$ degrees; K (canvas) $= 0.75$; $K' = 0.544$; $Q = 1.294$; $C = 0.04$; $N = 0.451$.

(1.) Substituting in formula (16):

$$t_1 - 59 = 68 \text{ degrees and } t_1 = 127 \text{ degrees.}$$

(2.) To find the coefficients of K and K' for 68 degrees temperature-difference, divide the total coefficients given by Tables 1 and 3 by 68. Thus:

$$\text{Coefficient of } K = \frac{(64.96 + 1.403 \times 14)}{68} = 124.$$

$$K' = \frac{(65.88 + 1.56 \times 14)}{68} = 1.29.$$

Then the new value of Q is:

$$Q' = 1.24\,K + 1.29\,K' = 1.632.$$

(3.) Substituting the value of Q' in formula (17), we have the corresponding value:

$$M' = 105 \text{ B.t.u.}$$

(4.) To find the corresponding surface temperature t_2', we have from formula (16):

$$t_2' - 59 = \frac{105}{1.44 \times 1.632} = 45 \text{ degrees, and } t_2' = 104 \text{ degrees.}$$

(5.) For a second approximation, repeat the processes as above. Thus, repeating method (2) for a surface temperature-difference of 45 degrees:

$$Q'' = 1.186\,K + 1.176\,K' = 1.529.$$

This value of Q'' gives by method (3),

$$M'' = 104.3 \text{ B.t.u.},$$

which is only 0.7 B.t.u. less than M'. It may therefore be accepted as final. This result, like those computed for bare pipes, is subject to a correction to meet the conditions of pipes in service.

In finding the conductivity C of a covering from test results, the values of M and t_1 (for saturated steam) are known, and the surface temperature t_2 can be computed by the method of approximation employed above. Substituting these values in formula (15) gives that of C.

Table 9 gives some representative tests of pipe coverings by the condensate method. From these data, the conductivity and economy of the various coverings can be computed for comparison by the methods just described.

TABLE 9.—CONDENSATE TESTS OF PIPE COVERINGS.

(Saturated Steam.)

Investigator.	Pipe diameter.	Covering.		Temperature.		Economy: Heat saved, as compared with bare pipe, per cent.
		Material.	Thickness.	Air.	Steam.	
	Inches.		mm.	Degs.	Degs.	
Pasquay	......	Silk waste	26.	15 C.	135 C.	85.6
Pasquay	......	Silk waste and air jacket	39	15 C.	135 C.	87.9
Brill	8	Hair felt	0.82	69 F.	348.3 F.	84.4
Brill	8	Mineral wool	1.3	58.3 F.	344.1 F.	81.4
Haacke	5	Kieselguhr *	1.0	76.1 F.	307 F.	83.6
Capper	......	Mica †	1.6	......	404.2 F.	89.3
Barrus	10	Asbesto-sponge felt	1¼	62.8 F.	364.8 F.	91.3
Barrus	10	Magnesia	1 3/16	66 F.	365.2 F.	89.0
Barrus	10	Asbestos, Navy brand	1¼	66.8 F.	365.2 F.	88.0
Jacobus	2	Remanit ‡	1.3	76.1 F.	306.6 F.	86.9
Jacobus	2	Asbestocel	1.07	77.2 F.	301.8 F.	77.2
Jacobus	2	Asbestos air cell	0.96	72.3 F.	303.3 F.	74.4
Jacobus	2	Asbestos fire felt	1.2	72.5 F.	307.4 F.	73.1
Jacobus	2	Magnesia	1.08	81.6 F.	310.9 F.	83.2
Bolan and Grieve	3	Slag wool	1.25	......	308.1 F.	79.4
Bolan and Grieve	3	Magnesia, sectional	1.5	......	308 1 F.	80.5
Zulauf	7.56	Snowdon's asbestos composition	2.28	83.3 F.	366.0 F.	89.2
Zulauf	7.56	Compressed cork, Frick's	2.76	85.1 F.	366 F.	88.6
Zulauf	7.56	Isolgurit §	2.28	85.1 F.	366 F.	83.6

* Fossil meal composition.
† Made in flat or curved mattresses enclosed between casings of wire netting.
‡ Braids or pads of carbonized silk; pads enclosed in a net work of fine wire.
§ In fusorial earth with fibers.

In Table 10, there are assembled some covering tests made by other than condensate methods. For example, Norton's tests were conducted on a pipe filled with oil, heated electrically by coils of wire immersed in it. The method of Benisch and Anderson was similar, except that the closed pipe contained air instead of oil. In Stott's tests, the pipe was closed and filled with air, but it was heated directly by insulated cables soldered to its ends.

Electrical tests, especially those by the direct method, give an accurate measurement of the heat passing through the covering, thus avoiding the uncertainty and possible error of the condensate method. The basic principle governing them is that, if R be the resistance of any conductor at a given temperature, the energy required to maintain it at that temperature is C^2R, in which C is the measured current. If C_b be the current required for the bare pipe and C_c that for the covered pipe, then the B.t.u. saved per hour by the covering are:

$$3.41\ R(C^2_b - C^2_c).$$

It should be observed, however, that, while such tests measure accurately the conductivity and economy, they have no relation to steam transmission, in which the heat loss is affected by the coefficient of heat emissivity of the steam, and, if the latter be superheated, by its velocity also.

A method of heat insulation which has received less attention than its merits, is the use of air jackets which encase the pipe with a film of air too thin for convection to be a serious factor. It is the entrapped air in their more or less cellular structure which produces the economy of most pipe covering materials. With regard to air jackets of sheet-tin or zinc, Russner* says:

"I have found in my experiments that, at a steam pressure of one atmosphere, the most favorable thickness of the air stratum or the distance between pipe and jacket, is about 15 mm. With such a distance I have succeeded in reducing the quantity of water of condensation 74 per cent., and, with two jackets, 85 per cent."

TABLE 10.—ELECTRICAL TESTS OF PIPE COVERINGS.

Investigator.	Pipe diameter.	Covering.		Temperatures deg. F.		B.t.u. lost per sq. ft. of pipe surface, per minute.	Economy: heat saved by covering, per ct.
		Material.	Thickness.	Air.	Pipe.		
	Inches.		Inches.				
Norton..........	4	Bare pipe....................	......	72.0	388.0	13.84	0.0
Norton..........	4	Nonpareil cork............	1.00	72.0	388.0	2.20	84.1
Norton..........	4	Manville high pressure.	1.25	72.0	388 0	2.38	82.8
Norton..........	4	Magnesia....................	1.12	72 0	388.0	2.45	82.3
Norton..........	4	Imperial asbestos........	1.12	72.0	388.0	2.49	82.0
Norton..........	4	Calcite......................	1.12	72.0	388.0	3.61	73.9
Stott..............	2	Bare pipe....................	......	84.2	370.4	13.08†	0.0
Stott..............	2	No. 2........................	1.5	84.2	370.4	1.672	87.2
Stott..............	2	No. 6........................	1.0	84.2	370.4	1.371	89.5
Stott..............	2	No. 19	2.0	84.2	370.4	1.463	88.8
Stott..............	2	No. 20........................	2.0	84.2	370.4	1.544	88.2
Stott..............	2	No. 21........................	2.0	84.2	370.4	1.562	88.0
Benisoh and	......	Bare pipe....................	......	......	356 0	10.25	0.0
Andersen	......	Kalorit........................	0.787	......	356.0	4.43	56.8

†Computed from Péclet's formulae by the author. Covering No. 2: granulated cork, moulded and baked; No. 6: nine layers of asbestos paper with granulated cork in between; No. 19: eighty-five per cent. carbonate of magnesia in two one-inch layers; No. 20: solid two-inch moulded sectional covering, composed of eighty-five per cent. magnesia, finished with canvas; No. 21: same form and composition as No. 20.

*"Power," December, 1896.

PANAMA COLLIERS *ULYSSES* AND *ACHILLES.*

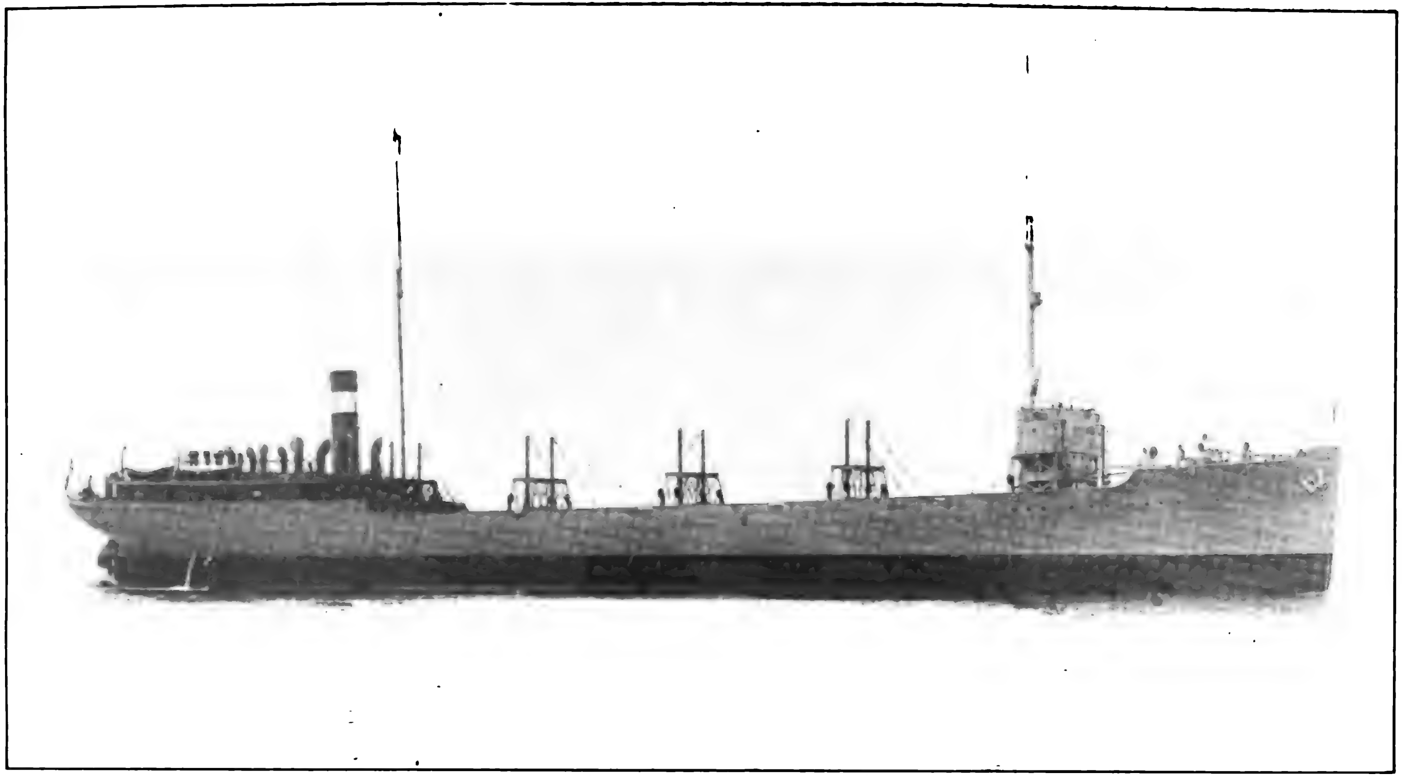
DESCRIPTION AND CONTRACT TRIAL PERFORMANCE.

By Henderson B. Gregory, Associate.

There have recently been completed and placed in commission two large colliers, the *Ulysses* and *Achilles,* for the Panama Canal.

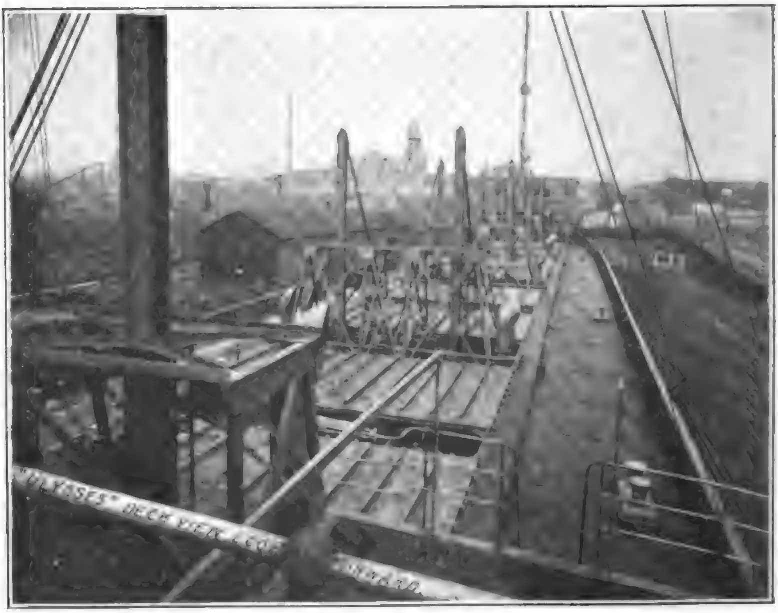
The vessels were authorized by an Act of Congress, approved June 23, 1913, and were built under contract by the Maryland Steel Company, of Sparrow's Point, Maryland, in accordance with plans and specifications prepared by the Navy Department. The contract price was $987,500.00, each, and the time of completion sixteen months from date of contract, April 9, 1914. They are twin-screw vessels, driven by reciprocating engines of 7,200 combined I.H.P., and designed for a speed of 14 knots at a displacement of about 19,585 tons, corresponding to the vessel complete, with all equipment, as usually carried on board in service; and with 12,000 tons of cargo coal, 1,200 tons of bunker coal, 20 tons of reserve feed water, 10 tons of drinking water, 75 tons of stores and crew. Provision is also made for carrying cargo fuel oil, when desired.

The construction of the vessels was in conformity with the Lloyd's Classification Society Rules and the United States Steamboat Inspection Rules, except as modified by the detail specifications. The vessels were inspected during construction by the Navy Department and representatives of the above-named organizations, and upon completion each vessel was furnished with a Certificate of Classification Lloyd's Register of Shipping for a vessel to Class 100–A–1, Lloyd's for American Government Service, Special Survey, and also a Certificate of Equipment that the vessel complies with the United States Steamboat Inspection Laws.



"ULYSSES" ON TRIAL TRIP, SPARROW'S POINT, MD., APRIL 10, 1915.

"ULYSSES," DECK VIEW LOOKING AFT.



"ULYSSES," DECK VIEW LOOKING FORWARD.

PRINCIPAL HULL DIMENSIONS.

Length over all, feet and inches	536–0
between perpendiculars, feet and inches	514–0
on load water line, feet and inches	514–0
Beam, extreme, feet and inches	65–2½
on load water line, feet and inches	65–2½
moulded, feet and inches	65–0
Mean designed draught, feet and inches	27–10
Displacement at designed draught, tons	19,414
per inch at L. W. L., tons	64.2
Area of immersed midship section at designed draught, square feet	1,780
L. W. L. plane at designed draught, square feet	26,600
Wetted surface at designed draught, square feet	47,320
Coefficient of fineness, block	0.724
midship section	0.984
load water line	0.8

GENERAL DESCRIPTION OF HULL.

The vessels are of steel, single-deck, self-trimming type, with topside ballast tanks and raised forecastle and poop deck. There are two masts and one smoke pipe.

Navigating Bridges.—There are two bridges located just abaft of the forecastle deck. The upper bridge is open and fitted with steering gear and searchlight platform, on which is mounted one searchlight. The lower bridge is semi-inclosed with steering platform in center, and chart house and bridge cabin, port and starboard, respectively. Below the lower bridge is the radio room.

Forecastle Deck.—This deck extends from the bow about 50 feet aft, and on it are located the anchor windlass and minor deck fittings.

Poop Deck.—The poop deck is about 175 feet long. On this deck is located a spacious deck house containing the officers' quarters, officers' and crew's galleys, pantries and bakery. There is also a small steam capstan aft of the deck house.

On top of the deck house are carried the boats.

Main Deck.—The main deck is continuous from the stem to the stern. It is a weather deck between the limits of the fore-

castle and poop decks. Under the forecastle are located the lamp room, paints and oils room, windlass engine, and crew's water closets and wash room. On the weather portion of the deck are the hold hatches and winches. Aft under the poop deck are the executive officer's and chief engineer's offices, crew's and firemen's wash rooms and water closets, crew's and chief petty officers' quarters, hospital and steering-gear room.

1st Platform.—Forward are a storeroom and spare berthing space, and chain lockers; and aft are located the engineer's workshop and stores, refrigerating and dynamo machinery, crew's berthing space and stores.

2d Platform.—The space forward is completely occupied by storerooms and chain lockers; and aft are located the refrigerating rooms, stores, fresh-water tanks and trimming tanks.

Hold.—From forward aft the hold contains the following: Forward trimming tank, pump room, holds for stores or cargo-oil, cargo-coal holds, steaming-coal bunkers and boiler room, engine room and the after trimming tank. Under the boiler and engine rooms are located the reserve feed and fresh-water tanks, respectively.

BOATS CARRIED.

The following small boats are carried in cradles on top of the deck house. The davits are of the Welin quadrant type:

3 28-foot metallic lifeboats;
1 30-foot self-righting and self-bailing motor lifeboat;
1 20-foot working boat;
1 14-foot punt;
Life rafts as necessary to provide for the remainder of the complement.

DECK MACHINERY.

Anchor Windlass.—An American Engineering Company's steam anchor windlass is located on the forecastle, with engine below on the main deck. The engine is of the vertical type with two steam cylinders 12 inches diameter by 14 inches stroke each.

Steering Engine.—The steering engine is located on the main deck aft. It is the American Engineering Company's horizontal type with two steam cylinders 12 inches diameter by 12 inches stroke each.

Capstan.—Aft of the deck house on the poop deck is an American Engineering Company's capstan operated by a horizontal steam engine with two cylinders 8 inches diameter by 8 inches stroke each.

Deck Winches.—There are eight steam deck winches on the weather portion of the main deck, for handling hatch covers, etc. The forward winch differs in design from the others, being equipped for serving the forward cargo holds. All winches were supplied by the American Engineering Company. They are of the horizontal double-cylinder type, cylinders 8 inches diameter by 8-inch stroke.

VENTILATION.

Cowls in number, location and size, as required for an efficient system of natural ventilation, are provided.

HEATING SYSTEM.

All living spaces, offices, bridge houses and quarters are steam heated in accordance with the usual practice.

STEAM FIRE-EXTINGUISHER SYSTEM.

A steam fire-extinguisher system is provided, consisting of 1½-inch independent connections to each hold and inner bottom tank. Steam is taken off the deck auxiliary steam line with a stop valve at each connection.

FIRE MAIN.

The fire main is supplied by a 6-inch connection from the fire and bilge pump, one of 8 inches from the ballast pump and

a 5-inch connection from the ash ejector pump. The fire main and topside ballast system are in common. Varying in diameter from 6 to 3½ inches, it extends fore and aft on the main deck, port and starboard, with hose valves conveniently located on the mains, and branches to the various parts of the vessel requiring hose plugs for fire protection and general purposes. All hose plugs are for 2½-inch standard fire hose.

DRAINAGE, PUMPING AND FLOODING SYSTEMS.

The customary drainage, pumping and flooding system is provided for the double-bottom tanks, topside ballast tanks, peak tanks and compartments requiring drainage. The tanks are filled and emptied by the pumps in the engine room and pump room forward. The topside ballast tanks drain overboard direct, through valves operated from the weather deck.

The topside ballast tanks on each side of the vessel are filled from the firemain, there being a branch with valve to each tank. The firemain has a branch forward for filling the forepeak tank.

The usual strainers are fitted in all drainage suction pipes to guard against coal dust, dirt, etc., choking the pipes, or entering the manifolds and pumps.

The double-bottom reserve feed tanks have connections only to fresh-water and feed pumps, and have filling pipes from the ship's side.

SANITARY SALT AND FRESH-WATER SYSTEMS.

The salt-water flushing system is supplied by the flushing, fire and bilge and distiller circulating pumps in the engine room. It supplies salt water to all water closets, galleys and to the firemen's and crew's lavatories and showers.

The fresh-water system is supplied by a gravity tank on top of the poop deck house, which is filled by a fresh-water pump located in the engine room. The fresh-water main supplies bath rooms and lavatories, pantries, galleys, scuttle butts, bakery, hospital, etc.

Individual water heaters are provided in bath rooms, etc., as required.

The ship's fresh-water tanks are located on the 2d platform aft. They have filling connections from the distilling apparatus and the ship's side, and suction connections to the fresh-water pump in the engine room.

CARGO FUEL-OIL SYSTEM.

The holds forward and inner bottom tanks, beneath the cargo coal holds, are fitted for carrying fuel oil when desired;

TABLE I—PUMPS AND CONNECTIONS—PANAMA COLLIERS ULYSSES & ACHILLES.

No.	Pumps	Size (Ins.)	Type	Suction Pipes from (Ins.)	Suction Pipes from	Discharge Pipes to (Ins.)	Discharge Pipes to	Location
2	Main Air	22 x 18	Attached	8	Condenser.	5 8	Feed Tank. Relief Valve.	Eng. Room.
2	Main Circulating	14" Nozzles	Vertical, Single, Centrifugal. 10" Cyl. x 9" Stroke, 39" Impeller.	14 10 10	Sea. Bilge. Cross-Connection.	14	Condenser.	Do.
2	Main Feed	14 x 9 x 24	Vertical, Piston, Double-Acting, Single, "Warren."	5 5	Feed Tank. Reserve Feed Tanks.	4	Main Feed Discharge.	Do.
1	Aux. Feed	Do.	Do.	5 5 5 2 5	Feed Tank. C. & R. Pipes from Ship's Side. Reserve Feed Tanks. Boiler Blow Pipe. Channelway.	4 4 4 2½	Main Feed Discharge. Aux. Feed Discharge. Reserve Feed Tanks. Overboard.	Do.
1	Aux. Boiler Feed	6 x 4 x 8	Do.	2 2	Feed Tank. Reserve Feed Tanks.	1½ 1½	Aux. Boiler. Aux. Feed Discharge.	Do.
1	Fire and Bilge	12 x 14 x 12	Do.	8 8 4 1½	Sea. Drainage. Bilge. Hose Connection.	6 7 2½ 2½ 4 3	Fire Main. Overboard. Hose Connection. Distillers. Flushing System. Water Service.	Do.
4	Bilge	5 x 18	Attached.	3½ 1½	Bilge. Hose Connection.	3	Overboard.	Do.
1	Flushing	8 x 8 x 12	Vertical, Piston, Double-Acting, Single, "Warren."	5	Sea.	4 4 3 2½	Fire Main. Flushing System. Water Service. Hose Connection.	Do.
1	Aux. Air	7½ x 14 x 10	Vertical, Single, Double-Acting, "Warren."	6	Aux. Condenser.	5	Feed Tank.	Do.
1	Aux. Circulating	6" Nozzles 6" x 6" Eng.	Centrifugal, Single Engine.	6	Sea.	6	Aux. Condenser.	Do.
1	Evaporator Feed	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, "Warren."	2 2 2	Sea. Overboard Disch. from Distillers. Main Condenser Disch.	2	Evaporators.	Do.
1	Distiller Fresh-Water	Do.	Do.	2½ 2½ 2½	Bottom of Distillers. Test Tank. Fresh Water Tanks.	2½ 1½ 2½ 2½	Fresh Water Tanks. Main Feed Tank. Reserve Feed-Water Tanks. Fresh-Water Main.	Do.
1	Distiller Circulating	8 x 8 x 12	Do.	3½	Sea.	2½ 2½ 1½	Distillers. Sanitary System. Refrigerating Machine.	Do.
1	Ash Ejector	14 x 8½ x 12	Vert. Piston, D.-A., Duplex, "Warren."	5	Sea.	5 5	Ash Ejector. Fire Main.	Boiler Room.
1	Ballast Pump	12 x 14 x 12	Do.	10 10 4	Ballast. Sea. After Peak Tank.	10 8 6	Overboard. Fire Main. Ballast Tanks.	Eng. Room.
2	Cargo Oil	14 x 12 x 18	Do.	6 8 5 4 6	Oil Holds. Sea. Double-Bottom Oil Tanks. Ford. Drainage Manifold. Deck. (Ship's Side).	6 10 4 5	Oil Holds. Deck. Overboard. Double-Bottom Oil Tanks.	Pump Room Ford.
1	Brine Circulating	4½ x 3½ x 4	Horizontal, Duplex, Double-Acting, "Worthington."	2½	Brine Return. (Surge Tank.)	1½	Brine Supply via Cooler.	Ice Machine Room.

the total capacity being about 757,500 gallons. The oil is handled by two large pumps, located in the pump room forward, with connections as given in Table I. There are two 8-inch filling pipes from the ship's side forward connected to the pumps' suctions, and four 6-inch combined gravity filling and deck discharge connections on the main deck forward.

MAIN ENGINES.

The main propelling machinery consists of two vertical, inverted, direct-acting, three-cylinder, triple-expansion engines, placed abreast, port and starboard, in a common compartment. The engines turn outboard when running ahead. They are designed to develop about 7,200 I.H.P. when running at 98 revolutions per minute, with a steam pressure at the H.P. valve chest of 195 pounds gage.

The sequence of cranks is H.P., L.P. and I.P., all crank angles being 120 degrees.

The cylinders are of cast iron, the H.P. alone being fitted with liners, and none are jacketed.

The housing and bedplates are of cast iron, the former of the inverted Y-frame type bolted to the cylinders and bedplates at top and bottom, respectively, and carrying the crosshead guides. The latter are of the box-section type, faced for the reception of the housings and supporting the main bearings, which are lined with white metal.

All pistons are of cast iron of the flat box-section design, fitted with packing rings and followers.

The piston rods and connecting rods are of forged steel. To the former are secured forged-steel crossheads of the double-slipper type, which in turn carry the crosshead slippers, of cast iron and lined with white metal, on both the go-ahead and backing faces.

The valve gear is Stephenson double-bar links, with linking-up screws for adjusting the cut-off. The H.P. and I.P. cylinders have one piston valve each, and the L.P. cylinder two

piston valves. All valves are provided with balance pistons. The eccentric rods and valve stems are of forged steel, actuated by cast-iron eccentrics secured to the crank shafts. The eccentric straps are of cast steel lined with white metal.

All piston rods and valve-stem stuffing boxes are metallic packed.

Reversing Gear.—Each engine is provided with a reversing engine of the direct-acting type connected through connecting rods to the reversing-shaft arms, the shaft in turn connecting to the links by arms and suspension rods. The gear is controlled by a floating lever operated at the working platform.

Turning Gear.—The customary turning gear is fitted on each engine, consisting of a single-cylinder engine with cylinder 6 inches diameter by 6 inches stroke, driving an intermediate worm shaft, which engages a worm wheel driving a shaft connected to a secondary worm driving a worm wheel attached to the main-engine crank shaft.

Lubricating Gear.—All working and moving parts of the main engines are efficiently lubricated by wick-feed distribution boxes located on the main cylinders and fitted with brass tubes leading to the various parts requiring lubrication.

Water Service.—A water service is provided for the main bearings, crank pins, crosshead guides and pins, and the thrust bearings.

Main Engine Data.

Diameter of H.P. cylinder, inches	27½
I.P. cylinder, inches	46
L.P. cylinder, inches	76
Stroke, inches	48
Piston rod, diameter, inches	7⅛
Connecting rod, diameter, crosshead end, inches	7¼
crank end, inches	8¼
length between centers, inches	111
crank ratio	1:4.625
Diameter, throttle valve, inches	9
1st receiver pipes (2), inches	9
2d receiver pipe, inches	18
Main exhaust pipe, inches	21 x 27¼

Valve settings from diagram:	H.P.		I.P.		L.P.	
Number of piston valves	1		1		2	
Diameter of valves, inches	$13\frac{1}{2}$		25		25	
Travel of valves, inches	9		9		9	
Inside or outside steam	Inside.		Outside.		Outside.	
	Top.	Bot.	Top.	Bot.	Top.	Bot.
Width of port, inches	3	3	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$
Steam opening, linear, inches	$2\frac{17}{32}$	$2\frac{21}{32}$	$2\frac{3}{8}$	$2\frac{9}{16}$	$2\frac{11}{32}$	$2\frac{15}{32}$
Steam opening, mean area, sq. ins.	80.05		155.17		294.6	
Exhaust opening, linear, inches	3	3	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$
Exhaust opening, area, sq. ins.	92.59		220		428.53	
Steam lap, inches	$1\frac{31}{32}$	$1\frac{27}{32}$	$2\frac{1}{8}$	$1\frac{15}{16}$	$2\frac{5}{32}$	$2\frac{1}{32}$
Exhaust lap, inches	$-\frac{3}{8}$	$-\frac{1}{16}$	$-\frac{5}{16}$	0	$-\frac{9}{32}$	$+\frac{1}{8}$
Steam lead, linear, inches	$\frac{3}{8}$	$\frac{1}{8}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{4}$
Cut-off, decimal of stroke	80.72	74.38	77.05	70.77	74.11	66.33

SHAFTING AND BEARINGS.

There are two lines of shafting of three sections each supported by proper bearings. The crank shafts are of the built-up type, and of solid forged steel throughout. The thrust and tail shafts are solid forged steel, the latter being composition cased in wake of the stern tube and bearings.

The crank and thrust shafts have disc couplings, forged solid with the shafts, and secured together by taper bolts. Each tail-shaft coupling consists of a forged steel sleeve, secured to the shaft by two feather-keys, forward of the sleeve are two half collars fitting a groove turned in the end of the shaft, all through bolted to the after coupling on the thrust shaft by parallel bolts.

Shaft Data.

Crank shafts, diameter, inches	$14\frac{7}{8}$
length, feet and inches	$23\text{–}7\frac{1}{2}$
number of cranks	3
throw of cranks, inches	24
angle between cranks, degrees	120
Crank pins, diameter, inches	15
length, inches	16
Coupling discs, diameter, inches	27
thickness, inches	$04\frac{1}{2}$
bolts, number each coupling	8
diameter at face of coupling, inches	$03\frac{1}{8}$

Thrust shafts, diameter, inches	14⅞
length, feet and inches	18–0⅞
collars, number	8
thickness, inches	02⅝
space between, inches	05¼
outside diameter, inches	24½
inside diameter, inches	15
bearing surface, sq. ins	2,212.88
Tail shafts, diameter, inches	15⅝
length, feet and inches	51–5⅞

Bearing Data.

Crank-shaft bearings (White metal lined):	
Number, each shaft	6
Diameter, inches	14⅞
Length, inches	17¼
Thrust bearings (White metal lined bearings and shoes):	
Diameter, inches	14⅞
Length, inches	16
Thrust shoes, number each	8
effective surface, sq. ins	1,548.8
Stern-tube bearings (Lignum vitae lined):	
Forward, diameter, inches	17⅛
length, inches	22
After, diameter, inches	17 3/16
length, inches	57¾
Strut bearings (Lignum vitae lined):	
Diameter, inches	17¼
Length, inches	64¾

PROPELLERS.

There are two true-screw three-bladed propellers of the adjustable-pitch and detached-blade type. The hubs are cast steel and the blades manganese bronze. The propellers have a taper fit on the tail shafts, and are secured by a key and nut.

Propeller Data.

Diameter, feet and inches	17–0
Pitch as set, feet and inches	17–0
Pitch	adjustable from 16 ft. to 18 ft.
Ratio of diameter to pitch	1
Projected area, square feet	81
Helicoidal area, square feet	94.29
Disc area, square feet	226.98
Height of lower tip of blade above keel, inches	18
Immersion of upper tip of blade at load draught, inches	114

MAIN CONDENSING APPARATUS.

Main Condenser.—There are two independent, cylindrical, surface, main condensers located one outboard of each main engine. The shells are steel plate, heads cast iron, tube sheets rolled brass, and tubes solid-drawn brass, No. 16 B.W.G. thick, ⅝-inch outside diameter and 10 feet 1 inch long between tube sheets. The tubes are secured in the tube sheets by the usual screw glands and packing. There are 3,213 tubes in each condenser, giving a total cooling surface in each of 5,301 square feet.

Particulars of the main air and circulating pumps are given in Table I.

Feed and Filter Tank.—A feed and filter tank is located in the forward port end of the engine room. The tank is divided into four compartments, with a filter chamber in the top of the second, third and fourth compartments.

ENGINE-ROOM AUXILIARIES.

Auxiliary Condenser.—There is one auxiliary condenser of similar construction to the main condensers. It contains 808 tubes and the cooling surface is 925.16 square feet.

Pumps.—The various engine-room pumps and their connections are enumerated in Table I.

Feed-Water Heater.—A Reilly multicoil feed-water heater, of 241.8 sq. ft. of H.S., is located in the port forward corner of the engine room, above the feed pumps. It is connected to the main feed line only. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each condenser connection, opening toward the condenser.

BOILERS AND APPURTENANCES.

Main Boilers.—There are three main boilers, of the horizontal, double-ended, Scotch type, arranged abreast in one compartment. Each furnace has an independent combustion

chamber. The general particulars of the boilers are given below.

Donkey Boiler.—A donkey boiler of the vertical, cylindrical, fire-tube type, is located in the boiler hatch, above the main boilers. The general characteristics of the boiler are given below.

Boiler Data.

	Main (each).	Donkey.
Working pressure, pounds gage	200	200
Mean diameter, feet and inches	16–00	7–00
Length of shell, feet and inches	22–01	....
Height, external, feet and inches	....	11–00
Thickness of shell, inches	01$\frac{17}{32}$	00$\frac{23}{32}$
Furnaces, type	Morison corrugated	Cylindrical
number	8	1
diameter, inside, feet and inches	3–05	6–04
thickness, inch	00$\frac{19}{32}$	$\frac{5}{8}$ (stayed)
Grates, length, feet and inches	5–06	circular
width, inches	41	6′ 3″ diameter.
Grate surface, square feet	150.33	30.68
Heating surface, square feet	6,545	1,046.54
Ratio grate surface to heating surface	1:43.54	1:34.11
Clear area through smoke pipe, square feet	71.72	4.91
to grate area	6.29:1	6.25:1
Tubes, material	charcoal iron, lap welded	
number, ordinary	708	196
stay	360	104
diameter, all, inches	02$\frac{1}{2}$	02
thickness, ordinary, B.W.G.	12	12
stay, inches	$\frac{5}{16}$ and $\frac{7}{32}$	$\frac{1}{4}$
length between tube sheets, ft. and ins.	7–10$\frac{1}{8}$	6–03$\frac{3}{4}$
Diameter, boiler stop valve, inches	08	03
safety valves, inches	04$\frac{1}{2}$ triple	02$\frac{1}{2}$
feed, stop and check valves, inches	03	01$\frac{1}{2}$
bottom-blow valves, inches	02$\frac{1}{2}$	01$\frac{1}{2}$
surface-blow valve, inches	01$\frac{1}{2}$	....

Uptakes and Smoke Pipes.—The main boilers are connected by suitable uptakes to one smoke pipe, 10 feet in diameter by 91 feet 1½ inches high above the lower grates. The uptakes and smoke pipes are double cased, with air space between casings. The smoke pipe is stayed by twelve wire-rope guys, arranged in two rows of six guys each.

On the front of the smoke pipe is a ladder, extending from the top of the boiler hatch to the top of the pipe, also a whistle and a siren located about 15 feet from the top of the pipe, and on the rear is a 12-inch escape pipe.

The donkey-boiler smoke pipe 30 inches in diameter, is led inside and to the top of the main smoke pipe.

FIRE-ROOM AUXILIARIES.

Forced-draft System.—The Howden closed ash pit, hot air, forced-draft system is installed for the main boilers. The air is supplied by two No. 9 Sturtevant, double-inlet, multivane fans, each driven by a vertical, single steam engine, with 5-inch steam cylinder by 4-inch stroke. One blower is located over each wing boiler. The air is drawn from the firerooms and delivered to the boilers through ducts, connected to heater boxes in the uptakes, where it is heated before delivery to the ash pits.

Ash Ejectors.—Two, above-water discharge, hydraulic, ash ejectors are installed, one in each fireroom. The pump for the apparatus is described in Table I.

Ash Hoist.—In addition to the ash ejectors mentioned in preceding paragraph, the starboard ventilator in the after fire room is arranged for hoisting ashes in a bucket by means of a 4½-inch by 4½-inch double steam engine made by the Hyde Steam Windlass Company.

MAIN STEAM PIPING.

The main steam piping is of seamless-drawn steel tubing. It is 9 inches in diameter and leads aft from the boilers to the engines in two lines, port and starboard. Steam is supplied through an 8-inch connection from each of the outboard boilers, and two 8-inch branches from the center boiler. The mains have a 7-inch cross-connection in the engine room. Stop valves are fitted near the forward engine-room bulkhead, at the boilers and in the cross-connection.

AUXILIARY STEAM PIPING.

The auxiliary steam piping is in two distinct circuits as given below. It is of seamless-drawn steel where subjected to boiler pressure, and copper for the loop in engine room, which is under reduced pressure.

From the main steam pipes in the engine room, with stop valves at the mains, 4-inch branches, starboard and port, form a common loop around the engine room. From these pipes branches are taken to the various engine-room auxiliaries, distilling apparatus, refrigerating and dynamo plants, steering engine, and capstan.

The fire-room auxiliary steam line is supplied from the engine room by a 5-inch branch. There is also a 3-inch connection from the donkey boiler to this line. From this pipe there are branch connections to the blowers, ash-ejector pump, etc., a 5-inch branch to deck machinery forward, the latter being led to the main deck, where it is carried forward along the deck, starboard of the coaling hatches, with branches to the winches and windlass engine. The deck piping is lap welded galvanized steel.

AUXILIARY EXHAUST PIPING.

The auxiliary exhaust pipe is of wrought iron, that exposed to the weather being galvanized. A main, 6 inches in diameter, is led throughout the machinery space, with connections from the various auxiliaries as required. The main is connected to the main and auxiliary condensers, there being a spring-relief valve at each connection opening toward the condensers.

The deck exhaust pipe, 4½ inches in diameter, is led along the port side of the main deck just outboard of the coaling hatches, with connections from the coaling winches and windlass engine, and down the boiler hatch to the auxiliary exhaust line in the fireroom.

BOILER FEED SYSTEM.

The main and auxiliary feed pumps, each, have a 5-inch suction from the 7-inch feed-tank suction pipe. Each pump has a 4-inch discharge connection discharging into the main and the

auxiliary feed lines, which are cross-connected in the engine room. The main feed line is 5 inches in diameter and discharges via the feed-water heater to the boilers. The auxiliary feed line is 4 inches diameter, but does not connect to the heater. All feed connections at the boilers are 3 inches in diameter. The feed discharge pipes are steel and the suction pipes copper.

INTERIOR COMMUNICATION.

The customary engine and fire-room telegraphs, gongs, voice tubes, etc., are fitted for transmitting orders and signaling to the machinery compartments and other parts of the vessels.

EVAPORATING AND DISTILLING PLANT.

Two evaporators and two distillers are provided, having a combined capacity of 8,000 and 4,000 gallons, respectively, in twenty-four hours. The former are located in the after end of the engine room, and the latter in the engine hatch. They are of the Reilly multicoil type, and operate in single effect.

Steam for the evaporator is taken off the auxiliary line, and the distiller fresh water is discharged by gravity to the fresh-water tanks and to the fresh-water filling manifold.

The distilling plant pumps are given in Table I.

ELECTRIC PLANT.

The dynamos are located on the 1st platform, port of the engine hatch. The installation consists of two horizontal, compound-wound, multipolar, direct-current, 25-kilowatt, Western Electric generators, each driven by a B. F. Sturtevant turbine. Each generator will deliver at normal load 200 ampères of current at 125 volts, when running 3,600 revolutions per minute.

REFRIGERATING PLANT.

A Brunswick ammonia compressor, of 2 tons' capacity, direct connected to a 5½ by 7-inch Wach's steam engine, is located on the 1st platform, port of the engine hatch, together with condenser, ammonia receiver and all appurtenances. The con-

denser circulating water is supplied by the distiller circulating pump in the engine room. The brine circulating pump and its connections are given in Table I.

The refrigerating rooms are located on the 2d platform just below and abaft the ice-machinery room.

ENGINEER'S WORKSHOP.

A spacious and complete workshop is provided on the 2d deck, off the engine hatch, starboard side. Its equipment includes the following machine tools, driven through shafting and belting by a 7½ H.P. 5-inch by 5-inch American Blower Company's vertical single engine:

1 16-inch swing by 2 feet 10-inch between centers engine lathe, Monarch Machine Company;
1 15-inch pillar shaper, Hendey-Norton;
1 22-inch swing upright drill, Barnes;
1 No. 4 Ransan grinder.

TRIALS.

The contracts required:

(*a*) A progressive trial over the measured-mile course at the Delaware Breakwater for standardizing the screws, extending from maximum speed (at least 14 knots) down to a speed of 8 knots; fourteen runs to be made over the course in order to adequately cover the range of speed desired.

(*b*) A full-speed and coal-consumption trial of twenty-four hours' duration in the open sea in deep water, at the highest speed attainable, which shall not be less than an average of 14 knots, determined from the official standardization curve of r.p.m. and speed. During the trial the air pressure in the ashpits shall not exceed an average of 1¼ inches of water, and the steam pressure at the H.P. valve chest shall not exceed 195 pounds above the atmosphere.

(*c*) Complete trials of the fuel-oil handling appliances and other apparatus, to demonstrate whether or not they fulfill the guarantees in the specifications.

Coal Consumption Guarantee.—The contractors guaranteed

that the coal consumption for all purposes on trial (*b*) shall not exceed 850 pounds per knot run, reduced to 14 knots speed in proportion to the fourth power of the revolutions.

PANAMA COLLIER ULYSSES
TABLE II—STANDARDIZATION TRIAL DATA—APRIL 14, 1915

No. of Run.	Time on Course		Speed in Knots.	R.P.M.			I.H.P.		
	Mins.	Secs.		Star. Engine.	Port Engine.	Mean.	Star. Engine.	Port Engine.	Total.
2	5	16.7	11.37	86.19	86.52	86.36	2397	2392	4789
3	4	15.0	14.12	83.84	84.35	84.20	2408	—	—
4	4	58.9	12.04	86.06	86.09	86.08	2398	—	—
Mean of Group.			12.91			85.21			—
5	3	49.5	15.69	97.73	96.70	97.22	3957	3700	7657
6	4	14.9	14.12	99.51	98.46	98.99	—	4442	—
7	3	50.2	15.64	99.34	98.27	98.81	4146	4593	8739
8	4	2.5	14.85	99.68	98.89	99.29	4127	4479	8606
9	4	0.7	14.96	99.37	98.26	98.82	4065	4375	8440
Mean of Group.			14.99			98.78			8588*
11	5	48.1	10.34	70.10	74.28	72.17	1615	1752	3367
12	4	55.7	12.17	70.55	71.60	71.08	1361	1683	3044
13	6	18.9	9.50	71.06	71.80	71.18	1414	1608	3022
Mean of Group.			11.05			71.38			3119
14	6	25.2	9.35	51.16	47.47	49.32	599	430	1029
15	8	59.4	6.67	53.34	54.06	53.70	593	761	1354
16	6	8.5	9.77	56.39	56.65	56.52	726	975	1701
Mean of Group.			8.12			53.31			1360

* Average of 7th, 8th & 9th Runs.

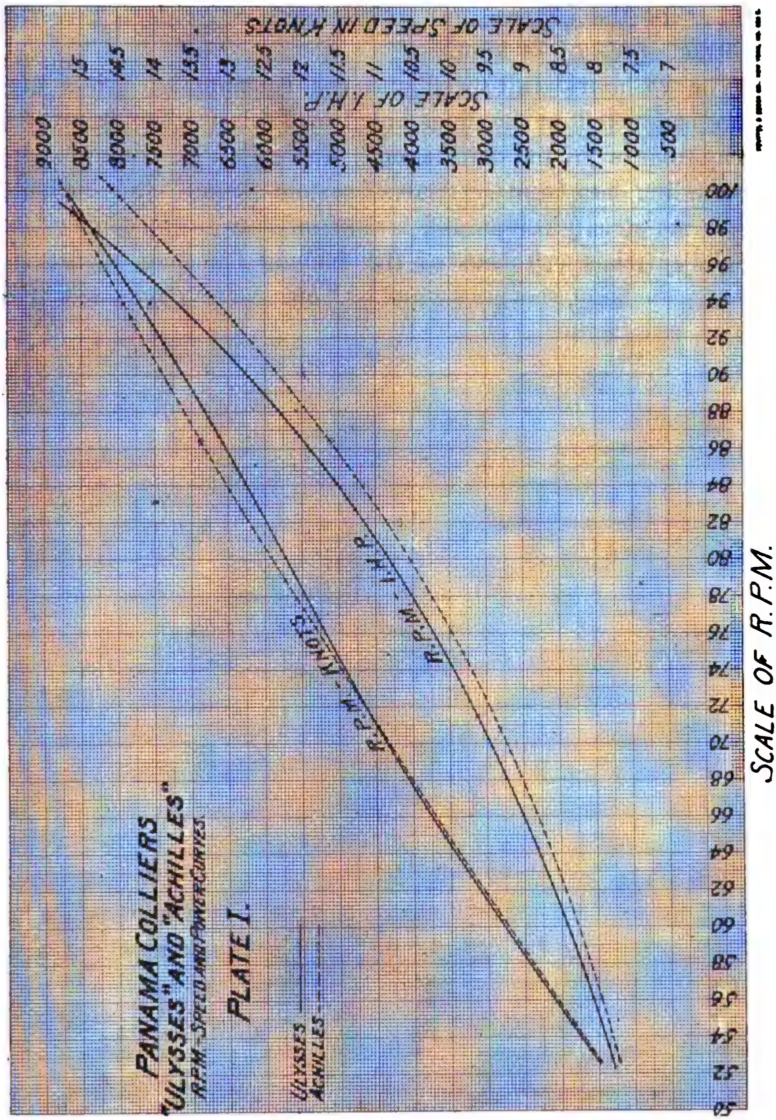
The standardization trial (*c*) of the *Ulysses* was run April 14, 1915, and that of the *Achilles* June 8, 1915. The mean displacement on the five high-speed runs was 19,375 tons and

PANAMA COLLIER ACHILLES
TABLE III—STANDARDIZATION TRIAL DATA—JUNE 8, 1915.

No. of Run.	Time on Course.		Speed in Knots.	R.P.M.			I.H.P.		
	Mins.	Secs.		Star. Engine.	Port Engine.	Mean.	Star. Engine.	Port Engine.	Total.
2	4	31.1	13.28	85.51	86.02	85.77	2327	2758	5085
3	4	38.5	12.98	86.20	85.76	85.98	2328	2659	4982
4	4	11.4	14.32	88.26	87.49	87.88	2612	2766	5378
Mean of Group.			13.37			86.40			5107
5	4	16.0	14.06	101.52	98.95	100.24	3942	4216	8158
6	3	36.5	16.63	101.58	99.22	100.40	3795	4119	7914
7	4	22.7	13.70	101.28	98.73	100.01	4195	3999	8194
8	3	34.8	16.76	101.69	99.11	100.40	4123	4049	8172
9	4	25.5	13.56	100.56	97.76	99.16	3983	3981	7964
Mean of Group.			15.23			100.13			8086
11	6	13.8	9.63	70.21	72.04	71.13	1295	1632	2927
12	4	58.3	12.07	69.86	71.55	70.71	1224	1569	2793
13	6	1.0	9.97	69.21	71.51	70.36	1215	1510	2725
Mean of Group.			10.94			70.73			2810
14	6	40.4	8.99	56.31	55.08	55.70	704	706	1409
15	7	25.1	8.09	54.58	53.15	53.87	670	644	1314
16	7	25.0	8.09	55.90	54.42	55.16	689	670	1359
Mean of Group.			8.32			54.66			1349

	…S	NEREUS	ORION	JASON
DATE OF TRIAL…	1913	SEPT. 3 & 4, 1913	JULY 10, 11 & 12, 1912	JUNE 18 & 19, 1913
DURATION, HOURS…		18	48	24
SPEED IN KNOTS…	5	14.575	14.467	14.322
DRAUGHT, MEAN O…		27-3¾	27-0½	27-0½
DISPLACEMENT, C…		18,838	18,800	18,862
PRESSURES (AVER…	…RT.	ST'B'D. / PORT	ST'B'D. / PORT	ST'B'D. / PORT
MAIN STEAM, AT…		197.5	195	199.6
		192.3	192	194.9
	.2	85.3 / 79.7	—	82.1 / 84.1
	.4	22.8 / 23.72	—	25.3 / 26.8
VACUUM, MAIN CO…	.02	26.44 / 26.52	—	25.84 / 26.14
BAROMETER, IN…		30.07	—	30.19
MAIN FEED, AT P…		—	—	—
AUXILIARY EXHA…		—	—	—
AIR PRESSURE…		0.985	1.0	1.334
MEAN EFFECTIVE…				
H.P. CYLINDER…	.3	81.6 / 86.1	—	85.5 / 82.8
I.P. "	.75	36.7 / 37.1	—	33.9 / 34.7
L.P. "	.2	9.8 / 11.1	—	9.7 / 10.7
REFERRED PRES…	.16	32.24 / 34.22	—	32.48 / 33.39
REVOLUTIONS OR…				
STARB'D. ENGINE…		98.82	94.76	96.04
PORT ENGINE…		99.27	95.24	94.35
MEAN…		99.07	95.0	95.2
MAIN CIRCULAT…	05	140.1 / 140.2	—	121.87 / 120.37
FEED PU…		13.85	—	11.35
FORCED DRAFT…	63	335.3 / 325.1	—	485.8 / 484.48
INDICATED HORSE…				
H.P. CYLINDER…	10	1020 / 1080	— / —	1091 / 1055
I.P. "	30	1308 / 1326	— / —	1284 / 1323
L.P. "	94	1017 / 1152	— / —	1002 / 1123
TOTAL…	34	3345 / 3558	3393 / 3550	3377 / 3501
BOTH EN…		6904	6943	6878
TEMPERATURES,…				
MAIN INJECTION…		75.3	—	63.1
OVERBOA…	.2	103.9 / 107.2	—	108.3 / 110.8
FEED TAN…		119.9	—	113.5
FEED WATER…		170.1	—	174.8
ENGINE ROOM…		88.4	—	87
FIRE ROOMS…		100.5	—	94.88
OUTSIDE AIR…		74.0	—	66.4
SMOKE PIPE…		—	—	—
COAL CONSUMPT…				
KIND…	…ER	NEW RIVER	POCAHONTAS	POCAHONTAS
QUALITY…	…D.	RUN OF MINE	FAIR	POOR
B.T.U. PER POU…		14,841	14,388	12,688
POUNDS PER K…	4	727.199	724.131	914.305
HO…		10,598.93	10,476	13,094.67
		1.535	1.509	1.904
		24.649	23.809	29.761
	2	0.573	0.554	0.692
MISCELLANEOUS…				
SLIP OF PROPE…	64	8.97 / 9.38	14.056 / 14.488	16.05 / 14.55
		9.18	14.272	15.3
GRATE SURFAC…		430	440	440
HEATING "		18,492	18,921	18,921
I.H.P. PER SQ. F…		16.056	15.78	15.682
SQ. FT. OF HEAT…		2.678	2.725	2.751
COOL…		1.304	1.586	1.55
KNOTS PER TO…		3.08	3.093	2.45
MAIN ENGINES,…	VERT.	3-CYL., 3-EXP., VERT.	3-CYL., 3-EXP., VERT.	3-CYL., 3-EXP., VERT.
	74	26 x 43½ x 74	27 x 46 x 76	27 x 46 x 76
		4	4	4

* I.H.P. MAIN EN…



19,293 tons, respectively. The data taken on these trials is given in Tables II and III, from which the curves, Plate I, were plotted. From the official r.p.m.-speed curves it was found to require a mean r.p.m. of both shafts of 92.5 and

90.81, for the *Ulysses* and *Achilles*, respectively, to give the contract speed of 14 knots.

The 24-hours' full-speed and coal-consumption trials (*b*) were conducted on the dates noted in Table IV, which gives a recapitulation of the data taken. Table IV also contains corresponding data from the trials of certain Naval Colliers, which, while not strictly comparable, are of interest in this connection.

Under trials (*c*) tests of the fuel oil handling apparatus, steering engine, anchor windlass, etc. were conducted and found satisfactory.

MANGANESE-BRONZE.

A DESCRIPTION OF ITS MANUFACTURE FROM VARIOUS SCRAP MATERIALS.

By Lieutenant J. B. Rhodes, U. S. Navy, Member.

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1. At the Washington Navy Yard the accumulation of a large quantity of miscellaneous non-ferrous scrap has led to the careful study of the metallurgical problems connected with the successful use of such materials. It is proposed to deal in this paper with the production of manganese-bronze ingots alone and to describe the materials used and the foundry practice, so that the experience gained here may be of value to others who are confronted with a similar problem.

2. The accumulation consisted chiefly of skimmings, turnings and trimmings, and obsolete or defective castings of compositions so doubtful that it was not considered advisable to use the materials directly in the production of castings. The yellow-metal scrap only is worth considering, although small amounts of red-metal scrap can be used to obtain the necessary tin. The following materials were available with composition approximately as shown:

1. Naval Brass: Copper 62 per cent., zinc 37 per cent., tin 1 per cent.
2. Cartridge-case Metal: Copper 68 per cent., zinc 31.6 per cent., nickel 0.4 per cent.
3. Manganese-bronze: Copper 59 per cent., zinc 41 per cent.
4. Commercial brass can be used in small quantities but should be avoided, as the lead content is too high.

3. Of these materials (1) naval brass and (2) cartridge-case metal can be and have been used in the manufacture of

cast naval brass, but the demand for cast naval brass has not been great enough to warrant holding scrap for use in that alloy alone, so it has been necessary to work out an economical and practical method for manufacturing manganese-bronze ingots.

4. The results of experiments during about six (6) months have shown that it is practicable to make high-grade ingots in an oil-fired Rockwell furnace of about two (2) tons' capacity. This has been accomplished in spite of the well-known prejudice against open-flame furnaces in the manufacture of non-ferrous alloys. Oxidation has been reduced to a very small amount by using wood scraps from pattern shop and salt. The bath is protected by the molten salt, and the wood insures a reducing rather than an oxidizing atmosphere in the furnace. The zinc losses are lower than is the case when no covering is used.

5. Before undertaking the manufacture of manganese-bronze itself a special hardener is made. This hardener is really the secret of the whole process, and although it can be made in any desired proportions, it has been found that the following is most satisfactory: 100 pounds copper, 25 pounds mild steel, 25 pounds (80 per cent.) ferro-manganese. The ferro-manganese and mild steel are melted and the copper (usually copper wire or other scrap) added as fast as the mixture will take it. Pots should be well stirred in the furnace. The alloy is best made using pots containing about 250 pounds each, and pouring these into a ladle for casting. Great care should be taken to skim the pots before pouring into the ladle. The hardener can be cast into ingots in green-sand open molds. After carefully skimming, both in the pot and in the ladle, it will be found that practically all of the carbon has been eliminated, as it separates from the alloy, floats on the top and is removed by the skimming. The resultant alloy will be quite clean and can be readily broken into pieces of any size. The alloy has a characteristic blue-gray color.

6. This hardener has proved most satisfactory as a means of

introducing the desired percentages of iron and manganese. Aluminum and tin are added at the end of the heat, as needed, to make up the desired percentages of these constituents.

7. In determining the proper amount of scrap for a charge the approximate analysis of scrap on hand must be known. It is necessary to consider the scrap as so many pounds of copper, tin and zinc, and sufficient accuracy will be found if we work out the mixture to contain 57 per cent. copper and add aluminum and tin as may be necessary. Suppose there is on hand a stock of scrap condenser tube containing, approximately, 70 per cent. copper, 29 per cent. zinc and 1 per cent. tin. In order to bring the copper content to 57 per cent. it will be necessary to use only about 78 pounds of scrap per 100 pounds of manganese-bronze. Zinc is added to reduce the percentage of copper.

8. The composition desired is as follows: Copper 57 per cent., zinc 40 per cent., iron 1.00 per cent., manganese, 0.75 per cent., aluminum 0.75 per cent., tin 0.50 per cent.

To obtain manganese 0.75 per cent., about 5 pounds of hardener should be used for every 100 pounds; this gives 2½ pounds copper, so that only enough scrap (condenser tube) should be added to bring copper content to 57 per cent., *i. e.*, 54.5 pounds of copper, which will be obtained from 78 pounds of scrap. This amount of scrap will carry 0.78 per cent. tin. In addition to these, one (1) pound of aluminum should be added for every 100 pounds of charge as computed. Aluminum, manganese and tin will be burned out to a slight extent.

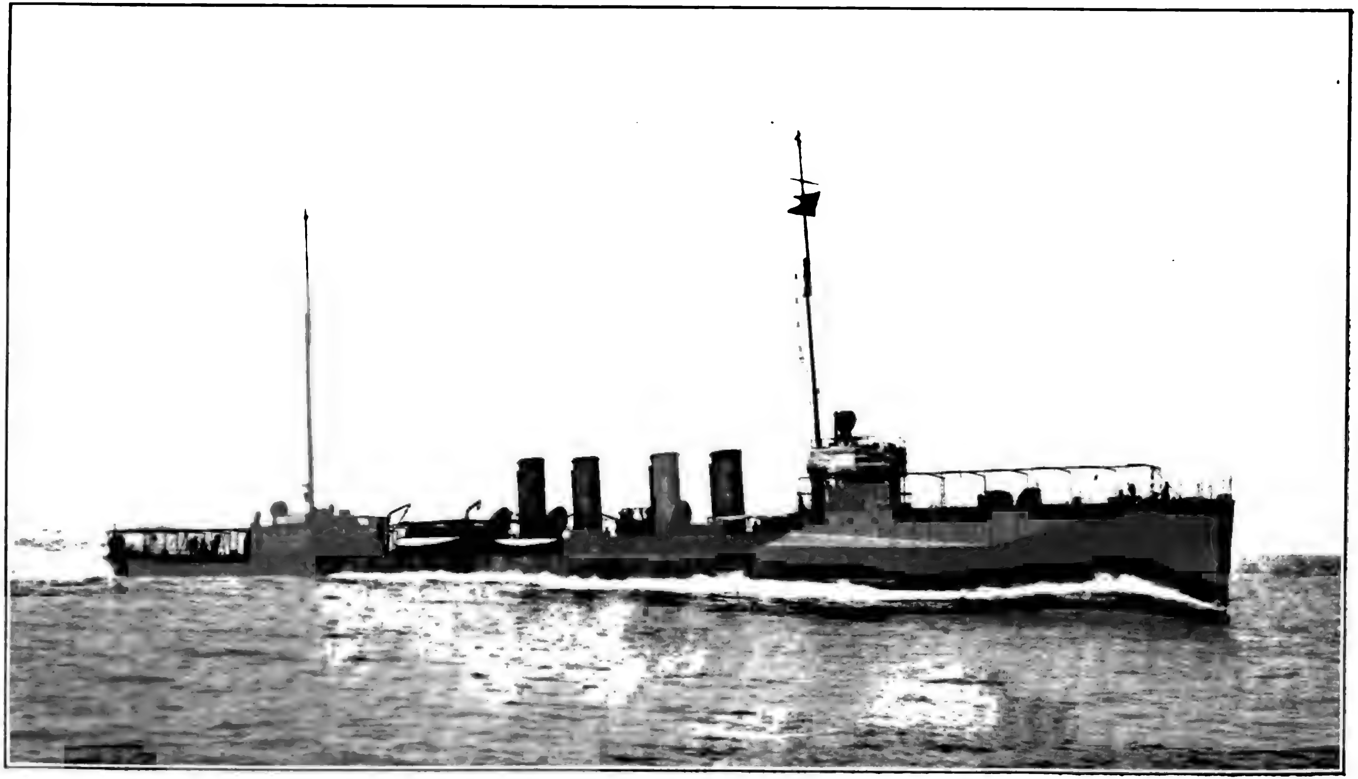
We now have 78 pounds condenser tube, 5 pounds hardener, 1 pound aluminum. To this should be added 16 pounds zinc to bring total to 100 pounds. This gives a zinc content of 38 per cent., assuming that there are no losses. About 5 to 8 pounds more should be added to make up for losses, which actually do occur, and the desired composition will finally be obtained.

9. After analysis of the heat the amount of zinc necessary to bring zinc up to 41 per cent. can be calculated, and this

amount should be added when remelting for pouring into castings. The best bronze, showing the highest tensile strength, is a bronze containing 41 per cent. zinc. If zinc drops to even 38 per cent., tensile strength is reduced and elongation is increased and a soft bronze obtained.

10. In melting in the oil furnace the most difficult scrap to melt should be charged first, although all but finals may be charged at once. As soon as melted the hardener should be added. In about half an hour charge the remaining scrap (if charge is not made all at same time) and continue the melt. After the heat is well up add zinc, then tin (if necessary) and finally aluminum; stir well and tap. Small ladles are used for pouring the ingots. Ingots are numbered to show the heat and turned into the store awaiting analysis. The cost of the method is high, on account of the labor in pouring and marking ingots, but, counting in furnace loss, labor, fuel and upkeep of furnace it is less than two cents per pound, so that scrap worth 7½ cents per pound can be converted into manganese-bronze to cost not over 10 cents per pound.

11. One of our heats gave 82,000 pounds' tensile strength and 28 per cent. elongation. Quite frequently 75,000 pounds' tensile strength and 20 per cent. elongation are obtained in sand castings. If high pouring temperatures are avoided and the metal is poured when it ceases to give off zinc fumes in large volume, it will be found that excellent values are obtained so long as the zinc content is kept up at or about 41 per cent.



Photograph taken by the New York Shipbuilding Company.

U. S. T. B. D. "Ericsson," on Trial Trip.

ERICSSON.

By W. H. A. LANGE, ASSOCIATE.

The *Ericsson*, Torpedo Boat Destroyer No. 56, is one of six destroyers of the same class authorized by an act of Congress approved August 22, 1912. These vessels were designated as Torpedo Boat Destroyers Nos. 51 to 56, inclusive.

The contract for the *Ericsson*, signed December 16, 1912, was awarded to the New York Shipbuilding Company, Camden, N. J., for $873,500, the vessel to be delivered twenty-four months after date of contract. Contract speed, 29 knots at about 1,090 tons displacement, main engines developing about 17,000 shaft horsepower.

The *Ericsson* is a twin-screw vessel equipped with Parsons turbines (main engines) in combination with a single reciprocating engine for cruising speeds.

PRINCIPAL HULL DATA.

Length on L.W.L., feet and inches	300–7
over all, feet and inches	305–3
Breadth on L.W.L., feet and inches	30–6
over guards, feet and inches	31–1
Draught, mean, feet and inches	9–9
Displacement, normal, tons	1,090
Tons per inch immersion at normal draught	14.50
Area immersed, midship section, square feet	204
Block coefficient	.427
Fuel-oil tanks, capacity, tons	309
Reserve feed-water tanks, capacity, tons	18
Additional, in cofferdam, tons	18
Fresh-water tanks, capacity, tons	17.5

The arrangement of machinery spaces, store rooms, officers' and crew's quarters, fuel-oil stowage tanks, reserve feed and fresh-water tanks and other minor compartments is, in gen-

eral, the same for all destroyers of this class. The propellers are located at the same frame space as those on the *Nicholson*, described in the previous issue of this Journal, but 10 inches farther out from the center line of the vessel.

MACHINERY.

The general arrangement of propelling engines and auxiliary machinery in engine rooms is shown on Plate I.

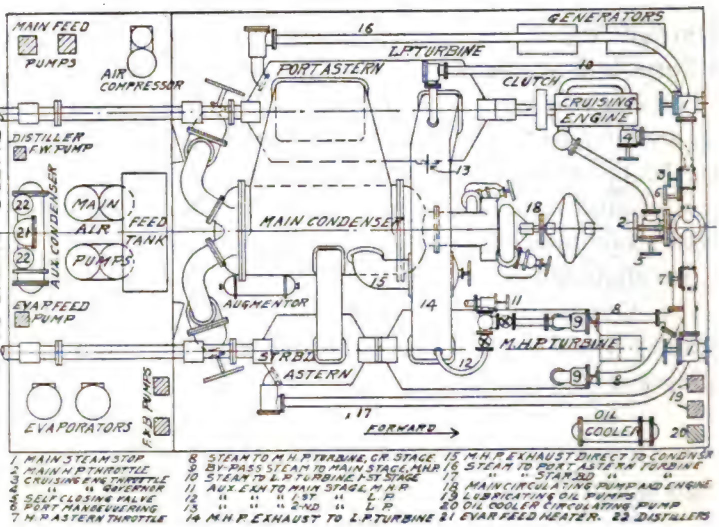


PLATE I. ENGINE ROOM

A vertical, inverted, direct-acting compound reciprocating engine, $\frac{16 \times 24}{18}$, with cranks at 180 degrees, is arranged forward of the low-pressure and astern turbine, on the port shaft; the main high-pressure ahead turbine and an independent astern turbine are mounted on the starboard shaft. The main engines are designed to develop 17,000 shaft horsepower at about 600 revolutions per minute when propelling the vessel at the contract speed of 29 knots.

A disconnecting clutch, of the Metten hydraulic type, is interposed between the cruising engine and the L.P. turbine for cruising speeds up to $15\frac{1}{2}$ or 16 knots. When speeds

greater than this are desired, the clutch is released without stopping the engine.

This clutch is operated with oil at a pressure of about 70 pounds per square inch. It is fully described by Mr. J. F. Metten, the inventor, in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXVI, No. 1, February, 1914.

A significant feature in connection with the arrangement of valves and piping to the reciprocating engine is, that, should the pressure of oil on the clutch diaphragm suddenly drop, the throttle of the reciprocating engine automatically closes instantly, thus preventing this engine from racing. Steam cannot be admitted to the astern turbine on the same shaft while the reciprocating engine is in use; neither can the throttle to reciprocating engine be opened while the port astern turbine throttle is open. By a slight movement of the handwheel on cruising-engine throttle, through an arc of 30 degrees, the astern throttle is locked in closed position and oil cock opened, applying full pressure to the clutch, which engages instantly, and to the governor valve of reciprocating engine, placing it in position for immediate action in case of emergency. The clutch is not again released until at the instant the throttle valve is closed. Steam can be admitted to the L.P. ahead turbine on the same shaft while the cruising engine is in use.

Metten clutch:	
Outside diameter, feet and inches	$3-10\frac{3}{4}$
Over all, inches	9
Disc, diameter, feet and inches	$3-7\frac{1}{4}$
Thickness, at center, inch	$0\frac{5}{8}$
periphery	$0\frac{5}{16}$
Intermediate shaft:	
Outside diameter, inches	8
Diameter hole through shaft, inches	5
at journal, inches	8
Length, feet and inches	$2-8\frac{1}{4}$

Ship coupling, intermediate shaft to L.P. rotor shaft, 8 steel pins $1\frac{3}{4}$ inches diameter on 12-inch pitch circle, fitted in steel bushings; allowable fore-and-aft movement, $\frac{1}{4}$-inch.

Line shaft:	
Two sections, outside diameter, inches	$8\frac{3}{8}$
Diameter hole through shaft, inches	5
at journals, inches	$8\frac{7}{8}$
Total length of both sections, feet and inches, about	34–7
Stern-tube shafts:	
Outside diameter, inches	$8\frac{3}{4}$
Diameter hole through shaft, inches	5
Composition casings at bearings, thickness, inch	$0\frac{7}{16}$
outside diameter, inches	$9\frac{5}{8}$
between bearings, thickness, inch	$0\frac{1}{8}$
Length of shafts, feet and inches	$27–6\frac{1}{4}$
Propeller shafts:	
Outside diameter, inches	$8\frac{7}{8}$
Diameter hole through shaft, inches	5
reduced at ends to diameter, inches	$2\frac{5}{8}$
Composition casings at bearings, thickness, inch	$0\frac{3}{8}$
outside diameter, inches	$9\frac{5}{8}$
Length of shafts, feet and inches	$21–0\frac{5}{16}$
Three line-shaft bearings, length, inches	12
Stern-tube forward bearing, length, feet and inches	2–4
after bearing, length, feet and inches	3–6
Propeller-shaft bearings, length, feet and inches	3–6
Propellers:	
Diameter, inches	$91\frac{1}{4}$
Pitch, inches	78
Projected area, one blade, square inches	1,341
three blades, square inches	4,024
Projected area ÷ disc area	.612
Developed area, one blade, square inches	1,500
three blades, square inches	4,500
Developed area ÷ disc area	.684
Specified r.p.m.	600
Specified shaft horsepower	17,000
Main condenser, tubes curved in a vertical plane and expanded in tube sheets at both ends:	
Cooling surface, square feet	10,800
Tubes, number	5,471
diameter, outside, inch	$0\frac{5}{8}$
thickness, inch	.049
Length between tube sheets, feet and inches	$12–0\frac{3}{4}$
Wet suction from the shell, diameter, inches	15
Dry suction from the shell, diameter, inches	15
Circulating water connections, injection, diameter, inches	30
Circulating water connections, outboard delivery, number and diameter	2 of 21

Main circulating pump, a Worthington 30-inch Bi-rotor centrifugal, volute, runners, diameter, inches	$27\frac{1}{2}$
Driven by a Terry steam turbine, type C.M.B. rotor, diam,, ins.,	36
Suction connections, number and diameter, inches	2 of 21
Discharge connection, diameter, inches	30
Suction connection from bilge, diameter, inches	7
Discharge connection to augmentor, diameter, inches	$5\frac{1}{2}$
Augmentor condenser, straight tubes:	
Cooling surface, square feet	320
Tubes, number	392
diameter, outside, inch	$0\frac{5}{8}$
thickness, inch	.049
Length between tube sheets, feet	5
Circulating water connections, diameter, inches	$5\frac{1}{2}$
Vapor inlet, diameter, inches	13
Discharge outlet, to water seal, diameter, inches	12
Auxiliary condenser, tubes curved in a vertical plane and expanded in tube sheets at both ends:	
Cooling surface, square feet	300
Tubes, number	316
diameter, outside, inch	$0\frac{5}{8}$
thickness, inch	.049
Length, between tube sheets, feet and inches	5–10
Circulating water connections, diameter, inches	4
Auxiliary exhaust nozzle, diameter, inches	7
Air-pump suction, nozzle diameter, inches	4
Feed and filter tank:	
Filter space capacity, gallons	285
Feed tank proper, capacity, gallons	510
Total capacity, gallons	795
Feed-water heaters, two, one in each fireroom, type, Schutte-Koerting, spirally corrugated tubes, $\frac{3}{16}$-inch film.	
Tubes, inner, inside diameter, inches	$1\frac{3}{8}$
thickness, inch	.083
outer, inside diameter, inch	$1\frac{13}{16}$
thickness, inch	.095
Heating surface, each heater, square feet	229
Lubricating oil supply, tanks, capacity, gallons	500
drain tank, capacity, gallons	200
settling tank, capacity, gallons	200
Evaporators, two, Reilly marine type:	
Heating surface, each, square feet	61.5
total, square feet	123
Specified evaporation, in 24 hours, gallons	3,750
Distillers, two, Reilly marine type:	
Cooling surface, each, square feet	18.45
total, square feet	36.9
Specified capacity in 24 hours, gallons	2,500

Evaporator feed-water heater, one, Navy type U tubes, heating surface, square feet	16.49
Oil coolor, one, Schütte-Koerting spirally-corrugated tube type, cooling surface, square feet	131.76
Reheater, in exhaust from cruising engine to H.P. turbine, straight tubes, heating surface, square feet	113.11
Fuel-oil heaters, four, two in each boiler room; type, Schutte-Koerting spirally corrugated tubes:	
Heating surface, each, square feet	11.08
total, square feet	44.32

Air compressor, one Westinghouse, 11 × 11 × 12, in after boiler room, for pneumatic tools and cleaning tubes.

Air compressor, one Ingersoll-Rand, in auxiliary machinery room for torpedoes. Capacity, 20 cubic feet at 2,500 pounds pressure.

Torsion meters, one on each shaft, in auxiliary machinery room; type, Gary-Cummings.

Generators, two (Westinghouse, steam-turbine driven), 25 kw., 125 volt D. C., in engine room.

Forced-draft blowers, two in each boiler room:	
Type, Keith fans, diameter, inches	32⅛
Terry steam turbines, rotor, diameter, inches	24

Boilers, four Thornycroft, water-tube, oil burning:	
Heating surface, each boiler, square feet	5,984
total, square feet	23,936
Working pressure, gage	265
Tubes, outside diameter, inches	1⅛
thickness, millimeters	109
number in each boiler	2,160
outside diameter, inches	1⅝
thickness, millimeters	120
number in each boiler	170
downcomers, two, diameter each, inches	
Steam drum, inside diameter, inches	42
Water drums, two, inside diameter, inches	19
Length, over all, including lagging, feet and inches	14–3½
Width, over all, including lagging, feet and inches	16–2
Height, over all, including lagging, feet and inches	13–6
Furnace volume, cubic feet	620
Area through uptakes, square feet	18.5
smoke pipe, square feet	18.5
$\frac{\text{Heating surface}}{\text{Furnace volume}}$, ratio	9.65
$\frac{\text{Smoke pipe area}}{\text{Furnace volume}}$, ratio	.03

Smoke pipes, 4, each 20 feet in height above deck line.

Oil Burners, Schutte-Koerting, 11 to each boiler.

No.	Purpose.	Pumps. Size, inches.	Type.
2	Main air.	$14 \times 28 \times 28 \times 18$	Twin, vertical, single acting.
1	Main circulating.	30 inch.	Centrifugal.
2	Main feed.	$15 \times 10 \times 16$	Vertical, simplex, double acting.
2	Auxiliary feed.	$15 \times 10 \times 16$	do.
3	Fire and bilge.	$7 \times 7 \times 12$	do.
1	Oil cooler.	$7 \times 7 \times 12$	do.
1	Evaporator feed.	$4\frac{1}{2} \times 6 \times 6$	do.
1	Distilller fresh water.	$3\frac{1}{2} \times 4 \times 4$	do.
1	Aux. condenser air and circulating.	$6 \times 8 \times 8 \times 7$	Horizontal combined air and water.
2	Lubricating oil.	$6 \times 5\frac{1}{2} \times 8$	Vertical, simplex, double acting.
2	Fuel-oil booster.	$6 \times 5\frac{1}{2} \times 8$	do.
4	Fuel-oil service.	$6 \times 3\frac{1}{2} \times 8$	Vertical, duplex, double acting.
2	Air compressor.	$11 \times 11 \times 12$	Westinghouse.

The main circulating pump is of the Worthington bi-rotor volute type, direct connected to a Terry steam turbine. All other pumps are of the Blake type.

Low Cruising.—Steam is admitted to the reciprocating engine, from which it exhausts, through the re-heater, to the cruising stage of the M.H.P. turbine, thence to the 1st stage of the low-pressure turbine and thence to the condenser. Clutch connected. Auxiliary exhaust steam is admitted to the main stage of the M.H.P. turbine.

High Cruising.—Clutch to reciprocating engine disconnected. Steam is admitted to the cruising stage of the M.H.P. turbine, thence to the 1st stage of the low-pressure turbine and thence to the condenser. Auxiliary exhaust steam is admitted to the 1st stage of the low-pressure turbine.

Full Speed Ahead.—Steam is admitted to the cruising stage of the M.H.P. turbine, with by-pass open to the main stage, thence to the first stage of the low-pressure turbine and thence to the condenser. Auxiliary exhaust steam is admitted to the second stage of the low-pressure turbine.

TRIALS—CONTRACT REQUIREMENTS.

(*a.*) A progressive trial over a measured-mile course not less than 40 fathoms in depth for standardizing the screws, extending from maximum speed down to a speed of 8 knots.

(*b.*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable; the average for the four hours to be not less than 29 knots an hour, to be determined by the average revolutions of the main shafts, according to the official standardization curve.

TABLE-I STANDARDIZATION. U.S.S. "ERICSSON" MAY 18, 1915.

NO. OF RUN	TIME ON COURSE		SPEED IN KNOTS	R. P. M.			S. H. P.			MEAN OF GROUP		
	MIN.	SEC.		STAR'BD	PORT	MEAN	STAR'BD	PORT	TOTAL	SPEED	R.P.M.	S.H.P.
1	8	18.8	7.22	126.65	124.87	125.76	86	91	177			
2	7	3.1	8.51	135.85	136.85	136.35	84	100	184	7.96	131.71	184
3	7	53.1	7.58	127.19	129.55	128.37	79	111	190			
4	5	2.8	11.89	196.03	191.64	193.84	291	289	580			
5	5	19.6	11.26	188.78	184.82	186.80	268	270	538	11.51	189.43	549
6	5	10.2	11.61	192.44	188.11	190.28	274	264	538			
7	3	44.1	16.06	262.97	267.47	265.22	780	815	1595			
8	3	50.5	15.62	263.03	269.74	266.39	699	887	1586	15.87	265.54	1561
9	3	42.7	16.17	261.11	267.19	264.15	694	781	1475			
10	3	00.3	19.97	345.87	343.83	344.85	1732	1802	3534			
11	2	56.3	20.42	343.09	342.07	342.58	1718	1709	3427	20.08	341.32	3400
12	3	4.5	19.51	338.38	332.13	335.26	1653	1558	3211			
13	2	26.0	24.66	437.24	436.37	436.18	3866	4068	7934			
14	2	31.0	23.84	436.27	437.16	436.72	3803	4049	7852	24.25	436.96	7850
15	2	26.1	24.64	438.12	437.06	437.59	3792	3968	7760			
16, 17		NOT USED										
18	2	16.5	26.37	503.60	512.54	508.07	*	*	*			
19	2	16.4	26.39	503.81	510.09	506.95	5825	6185	12010	26.45	508.50	12180
20	2	15.1	26.65	507.63	516.40	512.02	5869	6481	12350			
21	2	5.1	28.78	603.93	625.54	614.74	8103	9376	17479			
22	2	2.2	29.46	601.57	615.79	608.68	8257	9155	17412	29.06	610.75	17416
23	2	6.1	28.55	604.31	617.44	610.88	8220	9142	17362			
24	2	0.5	29.88	607.07	622.19	614.63	8332	9325	17657			
25	2	6.3	28.50	609.24	622.63	615.94	8513	9256	17769			
26	1	59.6	30.10	608.25	623.60	615.94	8349	9347	17696	29.33	616.98	17796
27	2	6.1	28.55	612.61	625.47	619.04	8598	8375	17973			
28	1	58.4	30.41	612.25	626.38	619.32	8555	9274	17829			

* NO S.H.P. OBTAINED ON THIS RUN.

(*c.*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service-cruising conditions.

(*d.*) A fuel-oil and water-consumption trial of four hours' duration in open sea at an average uniform speed of 15½ knots, as nearly as possible under conditions similar to the preceding trial, but with the cruising engine *connected* and in use.

Additional two-hour trial at about 15½ knots with main turbines only, oil and water consumption to be carefully measured.

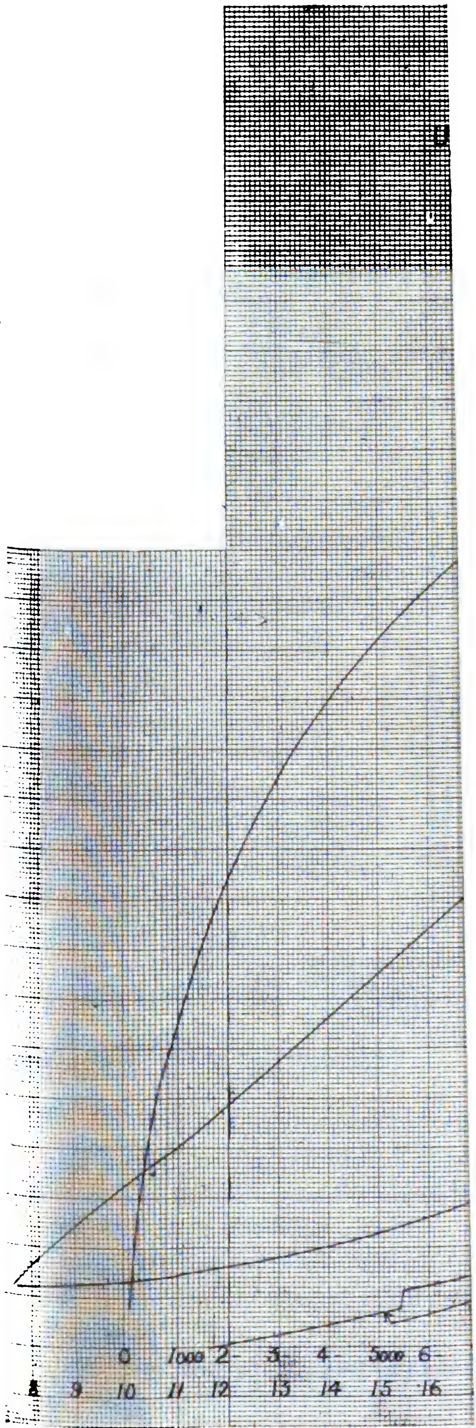

0 1000 2 3 4 5000 6
9 10 11 12 13 14 15 16

TABLE II. TRIAL DATA U.S.S. "ERICSSON."

	4 HOUR FULL POWER	4 HOUR 24 KNOT	4 HOUR 15½ KNOT	4 HOUR 12 KNOT	2 HOUR 15½ KNOT
DATE OF TRIAL	MAY 19, 15.	MAY 20, 15.	MAY 20, 15.	MAY 18, 15.	MAY 20, 15.
DISPLACEMENT (CORRESPONDING) TONS	1087.6	1097.8	1089.2	1096.3	1100.
DRAFT, MEAN, FEET AND INCHES	9-8¾	9-10¼	9-9⅜	9-9⅞	9-10⅜
SPEED, KNOTS	29.29	24.115	15.448	12.056	15.493
PROPELLERS, SLIP, PER CENT, STARBOARD	24.48	12.68	-6.18	1.04	3.25
PORT	27.23	13.79	16.75	9.35	9.90
MEAN	25.86	13.24	5.29	5.20	6.58
R.P.M., STARBOARD SHAFT	604.60	430.53	232.77	189.92	249.65
PORT "	627.43	436.09	288.01	207.32	268.05
MEAN	616.02	433.32	260.39	198.62	258.85
ENGINES IN OPERATION	TURBINES		TURBINES & CRUISING ENG.		TURBINES
S.H.P. AVERAGE, STARBOARD	8267	3740	326	245	622
PORT	8884	3410	773	175	509
TOTAL	17151	7150	1099	420	1131
I.H.P. AVERAGE, CRUISING ENGINE	—	—	770	185	—
NUMBER OF BOILERS IN USE	4	3	1	1	2
BURNERS USED (11 PER BOILER)	10.31	7.42	6.5	4	8
HEATING SURFACE USED, SQUARE FEET, TOTAL	23936	17952	5984	5984	11968
, PER S.H.P.	1.347	—	—	—	—
PRESSURES, AVERAGE					
MAIN STEAM, AT BOILERS GAGE	259.44	250	245.63	248.13	243.75
IN ENGINE ROOM "	248.25	241.5	241.5	239.88	238.65
CRUISING ENGINE, HIGH PRESS. CHEST "	—	—	208.75	91.25	—
L.P. RECEIVER "	—	—	91.	43.75	—
L.P. EXHAUST "	—	—	32.61	23.10	—
MAIN TURBINES, H.P., STEAM CHEST, ABS.	249.0	209.88	25.75	20.25	56.88
L.P. " "	25.0	12.50	2.56	2.38	3.88
H.P. 1ST. STAGE "	246.69	106.69	21.50	18.50	26.88
L.P. 2ND. " "	19.75	10.69	4.94	3.31	5.0
GLAND STEAM GAGE	3.98	40.0	4.63	3.9	4.0
AUXILIARY STEAM "	250.13	245.63	236.63	226.63	236.88
EXHAUST "	7.93	4.05	9.44	9.19	7.56
VACUUM, MAIN CONDENSER, INCHES MERCURY	28.18	29.20	27.94	28.88	29.21
BAROMETER " "	30.23	30.30	30.24	30.06	30.26
AIR IN FIREROOMS " WATER	6.38	4.52	3.69	2.63	1.83
FEED WATER AT HEATER OUTLET, GAGE	305.98	314.63	330.63	307.5	320.
IN FEED HEATER SHELL "	5.5	3.12	7.63	8.63	5.13
FORCED LUBRICATION SYSTEM "	10	10.03	9.75	10.11	10
FUEL OIL SERVICE PUMPS, DISCH. "	175.47	126.25	146.43	142.5	121.88
BOOSTER " " "					
PRESSURE TO BURNERS "	169.38	122.5	143.13	125.75	123.75
LUBR. OIL PRESSURE TO CLUTCH "	—	—	70	70	—
TEMPERATURES, FAHRENHEIT.					
OUTSIDE AIR	59.88	61.88	59.63	60.5	60.6
ENGINE ROOM	82.69	88.25	89.88	91.38	89.5
AUXILIARY ROOM	90.13	91.88	95.58	96.75	93.63
FIRE ROOMS	86.25	87.32	86.25	98.25	97.25
MAIN INJECTION	55.44	54.38	57.00	58.13	57.13
OUTBOARD DELIVERY	71.13	64.50	58.25	59.13	58.88
MAIN AIR PUMP DISCHARGE	82.91	77.63	73.13	91.25	81.13
FEED TANK	83.38	79.63	87.38	93.38	87.38
FEED WATER (FROM HEATERS)	173.26	192.00	207.25	218.75	200.25
OIL TO COOLER	107.50	102.25	95.75	96.0	90.38
OIL FROM COOLER	93.88	90.75	86.75	84.0	82.38
WATER FROM COOLER	58.0	58.38	60.50	60.75	61.63
OIL DRAINS FROM BEARINGS, AVERAGE	108.0	101.95	94.55	92.67	90.73
FUEL OIL TO HEATERS	68.57	69.38	72.0	81.0	73.75
FROM HEATERS TO BURNERS	134.54	134.05	148.57	122.25	137.25
SMOKEPIPE GASES (PYROMETER)	616.56	511.88	481.88	413.13	390.0
R.P.M. FORCED DRAFT BLOWERS	1685.70	1297.29	1323.13	944.38	1078.13
MAIN CIRCULATING PUMP	717.81	573.75	442.5	596.25	586.25
DOUBLE STROKES PER MINUTE, OF PUMPS					
MAIN AIR	27.57	27.88	34.0	32.13	28.38
MAIN FEED	31.51	28.25	11.88	11.0	12.75
FIRE AND BILGE	32.36	32.38	40.0	42.5	28.5
LUBRICATING OIL	46.25	40.50	49.75	39.5	36.0
OIL COOLER CIRCULATING	73.88	68.0	46.5	45.75	48.0
FUEL OIL SERVICE	25.88	12.69	11.25	4.25	16.0
BOOSTER	42.19	31.25	22.75	—	24.5
WATER CONSUMPTION, ALL MACHINERY	280145	132933	40575	30662	54317
" " ALL MACHY., PER S.H.P. MAIN ENGS.	15.762	17.40	28.275	42.286	37.589
COOLING SURFACE MAIN CONDNSR. SQ. FT. PER S.H.P.	.609	1.414	7.526	17.008	7.474
FUEL OIL CONSUMPTION, POUNDS PER HOUR	21856.4	9353.99	2906.77	1848.47	3182.07
" PER KNOT RUN	746.2	387.89	188.17	153.33	205.39
" " S.H.P. PER HR.	1.230	1.224	2.026	2.911	2.202
SPECIFIC GRAVITY	.8767	.8767	.8767	.8767	.8767
B.T.U. PER POUND	19474	19474	19474	19474	19474
POUNDS WATER EVAPORATED PER POUND OF OIL PER HOUR	12.818	14.211	13.369	16.588	17.070

(*e*.) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engine connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engine.

(*f*.) A fuel-oil and water-consumption trial of four hours'

duration in the open sea, with the cruising engine connected and in use, at an average speed of 12 knots, as nearly as possible.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 675 pounds at guaranteed maximum speed, 435 pounds at 24 knots, 200 pounds at 15½ knots, and 160 pounds at 12 knots, the consumption of fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,000 B.t.u. per pound of fuel oil.

INSPECTION NOTES.

By Captain F. W. Bartlett, U. S. Navy, Member.

FIBER.

There are two kinds of fiber, known to the trade as "hard" and "flexible."

Hard fiber is likewise known as "Vulcanized Fiber," "Hard Fiber," "Horn Fiber," etc., but the general name of "fiber" covers all of these and represents the material used so largely in the Naval service.

Soft fiber really means moist fiber. For this kind there is introduced in the bath liquids of various kinds to retain moisture in the product. Two of the common liquids are glycerine and solution of calcium chloride. Manufacturers have their secrets at this stage and claim better results than others in consequence. Beyond this the methods are alike, and the only other feature affecting the quality are selection of materials and care in manufacture.

Physical Qualities.—Fiber is unaffected by any of the ordinary solvents; it is not injured by immersion in alcohol, ether, ammonia, naphtha, turpentine or similar products; nor by any of the animal or other oils. It improves with age and is said to "season" as lumber does. It is believed that this improvement simply means lessened moisture. All fiber should season a month or more according to thickness.

It absorbs water very freely up to about 50 per cent. of its weight, and after being moist it warps and twists upon drying much the same as wood, although it will, after drying, resume its original thickness and qualities.

It is made only in sheets and tubes and cannot be molded.

Fiber will not melt under any circumstances and is not

readily burned, but at a very high temperature it chars and becomes brittle.

The tensile strength varies from ten thousand to twenty thousand pounds per square inch. Compression varies from forty thousand to sixty thousand pounds per square inch. Specific gravity is about 1.38.

Characteristics.—About the consistency of horn; good insulator; great mechanical strength; tough, tenacious, pliable, durable; capable of being sawed, turned, bent, embossed, drilled, threaded, cut, sheared, punched, etc., etc.

Sizes.—Sheets are approximately forty-five inches wide by seventy-two inches long and vary in thickness from five one-thousandths inch to two and one-half inches. Longer sheets may be furnished if desired.

Due to irregular shrinkage the sheets vary from forty-two inches to forty-eight inches wide and from sixty-six to seventy-two inches long. If needed of accurate dimensions they can be furnished to size at considerable greater expense. They should ordinarily be ordered about forty-five inches wide and about seventy inches long and the quantity in square feet paid for. Or payment could be made by weight, using twenty cubic inches to the pound.

Tubes are usually made with an inside diameter of ⅛ inch to 3 inches with any desired thickness of wall up to one-half inch and from two to three feet long.

Rods are furnished in random lengths up to 66 inches and in diameter from 3/32 inch to 2 inches.

Colors.—The color of the fiber depends upon the color of the paper of which it is made and there is no difference in quality due to the color. The common colors are red and black and a natural color approaching white. Manufacturers claim that uncolored fiber is best, as the coloring material used has solid particles, and prevents the perfect homogeneity necessary for best results. Many people are surprised that fiber is of any color but red. Uncolored fiber should always be ordered, except when colors are needed for ornaments.

Uses.—Fiber is an excellent insulator for electricity when dried and may be used for electrical purposes in all places that are always dry.

It is very useful also for condenser-tube packing, rollers, journals for light shaft gears of small sizes; likewise washers, brake bands, etc. It is particularly valuable in places where it is desired to deaden noises.

Method of Manufacture.—The best fiber is made from paper made from pure and specially selected cotton rags, and sometimes of cotton leavings, like waste. Cheaper fibers are made of cheaper material. This purity is necessary because this type of paper contains the greatest amount of cellulose that paper made by present methods contains. The need of the cellulose is so that it may be turned into a gelatine-like substance which acts as a glue and binder to the various thicknesses of paper. The paper is said in the trade, to be parchmentized. The process is usually carried out by the use of sulphuric acid. However, for fiber, zinc chloride is used instead of sulphuric acid and the paper is said to be "gelatinized," and fiber may be called a dried gelatinized cellulose.

The paper is made or bought in rolls of the trade width, and on its way to a pair of horizontal rolls is led through a solution of zinc chloride. This bath is only about six feet long and the paper dips into it and in that way becomes gelatinized. The paper is then led to the lower roll and is wound up on this roll, one thickness above another; the upper roll is comparatively light and merely rests on the top. Pressure is not particularly required as the fiber is made by the gelatinized surfaces of the paper sticking together and making practically a solid substance. If it were ideal it would be a concrete mass of gelatinized cellulose, but the paper carries this material and assists in making it stronger.

The rolls are hollow and heated by steam through the trunnions so that they are kept merely warm, not hot.

The rolling is continued until the required thickness is obtained. As it is found that this moist paper loses about one-

half of its thickness when dried, the material is rolled to about double the final thickness ordered. This shrinkage in thickness accounts for the great density and strength of the fiber. It is also found that this material becomes shorter than when rolled, so that it is always rolled about 20 per cent. wider than the final product is to be.

Finally, there is an annular roll of paper on the lower one of the rollers. When thick enough a horizontal cut is made of the paper on the lower roll and it is peeled off and flattened out.

The next operation is to clear the fiber of the zinc chloride, for two reasons. Zinc chloride will prevent using this material as an insulator. Also, zinc chloride is abstracted and used over and over as a question of economy.

There are many vats where this material is made; probably as many as one hundred, each one containing about the same quantity of water. The fiber is taken from the rolls and put in one vat after another to allow the zinc chloride to be entirely dissolved by the water, so that there may be none remaining in the final product. The length of time the fiber remains in each tank is a matter of experience. The water for these tanks travels in the opposite direction from the fiber. In other words, the purest water finally operates on the purest fiber. It is then pumped to the next tank and the water continues on, tank by tank, until the final tank, which is the one first receiving the fiber, becomes strongly saturated with zinc chloride. This is then pumped out and the water evaporated and the zinc chloride obtained for further use. The periods for changing the water depends on the results of experience.

This washing operation takes a long time. The operations of washing, drying, pressing and rolling for the varying thicknesses of the material take approximately the following lengths of time: ½ inch, 3 months; 1 inch, 6 months; 2 inches, 10 months.

This is the difficult feature of the business of making fiber, as once rolled to a definite thickness, this thickness cannot be changed, and if material in stock is of a certain thickness and

that thickness is never called for, there is an absolute loss of the entire process. In other words, it is necessary for a firm making fiber to make the different thicknesses on judgment of the market requirements. It is an absolute gamble as to whether they will ever sell different thicknesses or not, and a large sum of money must be kept idle to be ready for orders that may arrive, as it would be impossible to make the material of the thicker grades and fill any orders in reasonable time. With one firm, it is estimated that there was $1,000,000 worth of finished goods on hand.

After washing, the sheets are placed in dry houses. The idea in these dry houses is to imitate a warm summer day as nearly as possible, keeping them at a temperature of 90 degrees and have blowers arranged to change the air every forty-five seconds. When the fiber dries it curls up, shrinks and is very irregular in its shape.

After washing, the next operation is to press it flat by hydraulic pressure with steam-heated plates above and below. Those plates are of a temperature of 300 degrees, and the pressure on sheets 6 feet by 4 feet is 200 tons. Steam heat is used because fiber softens somewhat under heating, and the thicker the plate the longer the time required to be thus heated and flattened. When the operation is completed the fiber will remain flat. This heat and high pressure adds nothing to the density nor does it make the fiber thinner. The pressure put upon this material could not affect its density. This is simply a flattening operation after the drying.

The last operation is rolling or, as it is called in the trade, "calendering." The double horizontal rolls used weigh 5,000 pounds each and are held firmly with heavy screws at the ends. This is primarily done to make a better surface and is always part of the process. The thickness may be reduced somewhat by this rolling. As the material can flow a little, no harm is done to its qualities.

If close limits of thickness are required, sheets are picked

out slightly thicker than required and the calendering thins them to the proper amount.

Defects.—Much material is defective and its value and the labor expended is lost, as no use has been found for the waste due to faulty work, cuttings, drillings, etc., in spite of efforts made for years. Some defects are as follows and can be detected by observation:

(*a*) Iron in the paper that has escaped the close inspection of the rags and also the electric separator. This shows as a red coloration or rust spot. The iron, of course, ruins the fiber for electric work, and as this is the main use for fiber, the iron is closely watched.

(*b*) Blisters. These appear like blisters on wrought iron. The cause seems to be too uncertain for manufacturers to define accurately. It means that the layers of cloth are not glued together properly in places.

(*c*) Laminations at edges of plates are easily detected.

Stowage.—Fiber is stowed away in dry storehouses. The storehouses are not heated to any extent, although heat softens the fiber somewhat; but if heat is continued until all the water is gone from the fiber, it becomes entirely too brittle. Intense cold also produces brittleness. The stored material may be sold from storehouse or may be taken to the shops to be worked into shapes desired.

Manufacture of Tubes.—Tubes are made in the same manner, except that the paper is rolled on steel mandrels of proper inside diameter. As the paper is rolled on the mandrels it is carefully calipered, and when the desired thicknesses are obtained the paper is cut and the loose end rolled tight. The roll and mandrel combined are soaked in the same manner as mentioned before, and at the end of the soaking the mandrels are removed and new ones inserted. These new mandrels are the ones that positively settle the inside diameter of the tube, as the fiber dries closely to the mandrel, and as this fiber dries it shrinks to one-half the thickness as put on, making a very hard substance with a clear and accurately dimensioned hole

in it. Later on, after calipering to size, these mandrels are forced out. By driving square mandrels into the wet tubes, approximately square tubes are made.

Rods.—Rods used of this material are made from flat sheets. Sheets are sawed to size and turned round just as wood would be treated.

Manufacture of Various Articles.—Except for certain articles with holes lengthwise which are made from tubes, articles manufactured from fiber are made from sheet material; the nearest thickness is taken, and then it is a question of working this material, just as wood would be worked. It is astonishing to note the multitude of articles made from fiber and how its use is extending in all directions for purposes never thought of before. It is particularly advantageous where lack of noise is to be considered and where it is necessary to work in various acids and alkalies.

Threads.—It is difficult to secure good threads on fiber in many cases, and this is the principal fault that has been found with this material, as manufacturers expect more of material in this line than can be easily accomplished. It will be remembered that the paper is put on the roller in layers, so that when flattened out it leaves a definite lengthwise fiber of paper, like the grain in wood. When the thread is cut, therefore, on a round rod, on two sides diametrically opposite, there will be found the extension of these paper layers, and in threaded work, it is often possible to discover by close inspection just which way the grain runs. It is these paper edges, or where the percentage of the paper is greatest, that cause the difficulty in cutting the threads, and special care should be taken in this kind of work. Excellent threads are made repeatedly by manufacturers, but the details of care required are not generally known, so that trouble may be experienced at navy yards and aboard ship from endeavoring to thread this material. Manufacturers claim that if poor threading follows, it is due to poor material to a large extent. Always order "uncolored horn" fiber when difficult threading is to be done.

Care in Working Material.—It is important that this material should be worked dry, as it is very sensitive to moisture. It must not be too dry or it will be too brittle. However, if kept in a very dry storeroom and found brittle, the material may be placed out of doors under cover and in a few days the proper amount of moisture will be absorbed. Separate the sheets to let the air get at them. The exact amount of dryness would have to come by experience. It should be kept as dry as it can be worked properly, if extreme accuracy is required. Fiber is like wood. It will shrink sideways of the grain and not endways. For instance, if a ring of this material is cut when it is wet, the next day it will be found out of round, and holes drilled accurately one day will be considerably out of place the next. Articles have been sent to users of this material and returned with the complaint that they were too large and would not fit. This is due to their becoming moist, as it is known that this material will swell, as it absorbs about 50 per cent. of its weight and swells about 30 per cent. if thoroughly wet. If a small article is found to be too large to go into place, it should be dried for a few days and it will be satisfactory.

Condenser Ferrules.—In the Navy, fiber has been extensively used for condenser ferrules and with varying reports of success. Some claim the very best of results and some claim almost disastrous results. It is believed that this is largely due to the fact that the ferrules were not of the right size to begin with. It has been found from experience that the annular spaces around the condenser tubes vary considerably, depending on where the condensers were made. It is also found that in an old condenser where the threads have been retapped there is a great irregularity, and difficulty has been experienced in putting in corset lacing with tools that worked well with the ordinary annular space. If the fiber is ordered to a size to just fill this annular space, it will then swell after being set up and wet to about 30 per cent., if free, so that it would entirely fill

diametrically, the space, and will make a tight joint, when the gland simply makes a tight joint at the end.

If the ferrule is quite small so that it cannot soak up enough moisture to swell diametrically and entirely fill the space, the end pressure of the gland cannot accomplish the desired result and make it swell throughout its length diametrically. The gland may make a tight fit in its proximity, but the whole length cannot ever be forced by the gland to the proper fit. Therefore, it is advisable for those using these ferrules to see that they are a fairly tight fit when dry. Any size may be ordered and made with extreme accuracy, but the ordinary stock ferrule may not fit all spaces.

ELECTRIC HIGH-PRESSURE FIBER OR BAKELITE-DILECTO.

This material is coming into extensive use in the Navy for wireless work. It is capable of extreme resistance to electricity.

Method of Manufacture.—This is along the same general lines as with fiber, but the quality of the paper is not so important, as the paper in this case mechanically picks up in the bath bakelite, which acts as a glue, whereas in the other case the paper had to be capable of gelatination and of itself glue the layers together. In this case the bakelite is the important thing and is used as a thick liquid in the bath. The paper simply makes the product less brittle than it would otherwise be and adds to its strength. Sometimes canvas is used instead of paper where great mechanical strength is required.

In making this material the object is to retain the chemical, so that no washing follows. The material is rolled to the required thickness, layer upon layer, and dried, and is subjected to heat and pressure for flattening. The same press flattens this material that flattens the fiber. After this flattening process the material is calendered. It is then ready for use.

The manufacture of this material is rapid, and orders may be taken for prompt delivery. There is no question involved of a large idle capital, as that in fiber, nor of prospective cus-

tomers. The reason this material is so expensive, is on account of the cost of the expensive material used. The main cost is for bakelite.

Bakelite.—Bakelite is a secret material, and different companies have been subsidized to use it in the manufacture of insulations. It may be used alone and cast or pressed to the desired shape, but in this form it is very brittle and it is difficult to do anything with it, so that strength and durability can only be obtained by using paper or canvas as carriers of bakelite.

MAGNESIA PIPE AND BOILER COVERING.

Source.—Magnesia for pipe and boiler covering is made from limestone or dolomite containing as much magnesium carbonate as can be found. One firm has rock of this kind containing about 40 per cent. of magnesia. The works are generally at the deposit, to save freight, and a cheap method of quarrying and conveying the rock is devised.

Calcining and Slacking.—The rock is a mixture of calcium carbonate and magnesium carbonate, from which the calcium must be removed. The rock is first calcined, using charcoal as the fuel. This is done in the usual way for making limestone. The resulting rock is the combined oxides of calcium and magnesium. In other words, the carbonate has been changed to the oxide for both elements. The carbonic acid gas thus produced is used later. The calcined rock is next mixed with water in special rotating vessels where the lime is slaked. All impurities remain in the vessel and all the lime is carried away by the overflowing water during the operation. The oxides are now in a state of suspension in the water, the percentage of solid material being very small. The later processes consist in a juggling of " states of suspension" and " of solution" to separate the calcium and the magnesia.

Carbonating.—The liquid is pumped into vertical metal cylinders of proper size (in one case noted, about five feet in diameter and about fifteen feet high), which have, leading from the top head to near the bottom, pipes for the entrance

of CO_2 derived, as noted above, from the calcining. This gas is pumped from the calcining kilns and returned to these cylinders under a pressure of about 70 pounds. It rises through the liquid, being absorbed by the oxides in suspension in the water, and reconverts them into carbonates. But this operation produces an important difference in the condition of the two carbonates; after the operation the calcium carbonate is in suspension and the magnesium carbonate is in solution.

Elimination of Calcium Carbonate.—The next step is the separation of the materials. The separation is achieved by a system of filtration. A series of cells of wood, perforated by small holes and lined with cotton canvas or cloth, act as filters. The liquid enters the interiors of these cells. The water passes out, carrying with it the material in solution (magnesium carbonate) and the material in suspension (calcium carbonate) is caught in the meshes of the cloth as a paste. Frequently the cells have to be opened up and the cloths stripped from the frames for the removal of the calcium carbonate. The cloths, after cleaning, are used again for filtering. This is the end of the calcium carbonate as far as this process is concerned.

Final Process.—The water containing the weak solution of magnesium carbonate is next put into other vertical cylinders of about the same size as above and is there heated by steam to about 190 degrees F. At this temperature the magnesium carbonate becomes again in suspension instead of in solution. The liquid is then pumped to open vats lined with thin cloth and most of the water is drained off, just enough remaining to allow the mixture to be pumped to the molds.

Molding, etc.—The molds are made on the same principle as the cells described above, with the same size of holes and the same kind of cloth. Many cells are placed beside each other and pressed together firmly and the liquid is allowed to enter them all at once. When all are filled so that no more liquid flows from the openings, the cells are opened. The material

that was in suspension has been retained by the filtering process and the mold is filled with this magnesium carbonate.

This substance is used for the drug trade, and is later cut up into ounce weights by a band saw.

The magnesium carbonate used for pipe or boiler covering has to be mixed with some other substance so that the resulting mixture may not be so brittle. Asbestos fiber is used for this purpose. The mixing occurs in special vats by special apparatus.

The molds for pipe covering and for valves, etc., are made in the same general manner except that cores are made as in foundry work and the outside of the cells correspond to the shape desired. This work is generally done in individual presses.

After the molds are made they are dried. This takes about six days in special extensive ovens. Later the molds are treated as if they were made of wood in as far as the methods of completion of shape are concerned. Planers doing exactly the kind of work done by wood planers shape the outsides and insides. The material is cut to exact length by saws. Later thin canvas is pasted on the pipe and valve coverings and the completed product is shipped.

SMALL SHAFTS.

A certain firm makes a specialty of small shafting, particularly small crank shafts. Inasmuch as it does not make the material, the firm's financial success must depend entirely upon its special methods in shaping the material, which the consumer might purchase from the firm's competitor who makes the material and likewise makes small crank shafts.

Certain special appliances used by this firm are noted here.

Pressing Shafts.—This firm's output of very small shafts competes with drop-forged shafts, but a press is used in manufacture in place of the drop forging. The usual method of drop forging is cheaper, but it is contended that the pressed

material is in better and more uniform condition than that made by the drop-forging method; the suddenness of the blow in drop forging failing to allow the metal time to be thoroughly worked and to shape itself without leaving portions in a constricted condition.

Metal Saws.—A special type of circular saw seems to do unusual execution and to cost less for replacement of blades than other saws. The circular disc has on its outer circumference, staggered enlargements on either side to receive short cutting tools. The grooves for receiving these tools are made with extreme accuracy and measured to about one-half thousandth of an inch. The tools themselves are of bar steel ground accurately to fit the grooves. The grooves are made long enough for a tool 4 inches long, and when the tool is first put in place it is driven into position with considerable force from a hammer. There is no wedge of any kind to hold the cutting tool in the disc, the friction due to the close fit being sufficient to hold it securely. It would seem at first that this is impossible, but it must be remembered that pressure against the cutting tool is practically all tangential. The small amount of force tending to drive it radially is successfully resisted by the tightness of the driven fit. These cutting tools, when dull, can be resharpened; and it is found that they can be utilized in this way until the original 4 inches is reduced to about 1½ inches, so that only about ¾ inch is left for the fit in the wedge. It is found that even this short distance is ample, and is sufficient to keep the tool from moving radially. The groove for the tool is out of the middle radial plane of the disc a slight distance, so that the outside cutting edge on either side of the disc is slightly beyond the disc, giving clearance so that the disc will never rub against the work. The cutting tool is also sloped back a little for clearance.

These saws have been supplied to the trade for many years, and it is seldom that a new disc is required, but new cutting tools are constantly furnished. It is found that the fit is so accurate and the wear in the groove is so imperceptible that

for nine years not a complaint has been received of the fit of these cutting tools when supplied.

This is a remarkably clean-cutting saw, and the cutting tools produce a cut about ¾ inch wide for the smaller sizes and wider cuts for larger sizes. The saw is used for sawing off lengths of shafting and for any other sawing. In some cases, where short lengths are required, two or more saws are placed on the same mandrel with distance pieces between them, and all cut at once. These saws are particularly valuable for working on the crank shafts. The webs are forged solid and a portion from the web has to be removed. No method better than drilling has yet been discovered for the line nearest the pin. In some cases a single large hole is drilled, then two saws are arranged to cut in to the edge of this central hole, leaving only slight fins on either side which are easily broken off. This method of removing material from between the webs is a steady operation and is much better than the method of having the webs swing around in the lathe to receive a sudden jolt as the shaft strikes the fixed tool.

Also, the double saw cleans up the outside of the webs in one operation. Sometimes one cut is sufficient to take off the extra material, but when more are necessary a special saw is used. This saw is designed on the same general principle as the one mentioned above. The cutting tools are arranged in sets of three, staggered longitudinally across the rim of the disc. Thus during all parts of the cutting one cutting tool takes off its width, and the next one is just outside of it and takes off another width, and a third one takes still another width. In this way the sets of three cutting tools combined will take off a width of an inch or more on each side of the web. This saw produces most remarkable results in the saving of labor and cost.

When the saws are dulled, they are sharpened by revolving them against revolving cutting wheels of emery or other abrading material.

Lathes.—Another special feature is a type of lathe used.

This lathe has a face plate at either end, both ends being driven by power which is carried along a square shaft from one end of the lathe to the other. The driving is done by pinions on this shaft through gear wheels on the head and tail face plates. These two face plates have holders on them for gripping the shafts, and no centers are used. The shaft is gripped by the holders in a central position. There are circumferential and radial adjustments for the holders, and accurately marked scales. After the shafting is gripped it is centered by the scale marks, so that the main bearings may be turned. To turn the crank pins the holders at each end are shifted a distance equal to the throw of the crank, as measured on the scales. This produces a remarkably quick shaft centering, and is absolutely accurate as long as the lathes are in good condition. As certain shafts made by this firm have as many as five or six cranks, the rapidity of this operation is extremely valuable. In addition to the radial adjustment the circumferential adjustment allows the shifting of the shaft the required angular distance between the axes of the several pins, by using the circumferential scales. This also is rapid and accurate.

There is a third support for the shaft near its middle length. This support has the appearance of a steady rest and does execute that function. But there is an annular wheel within the outer casting which circumscribes and grips the shaft, and this annular wheel is driven through gearing by the square shaft above referred to. The square shaft allows this central support to be moved as needed. With this additional driving support there is little danger of sagging of the work. Also, besides the cutting tools at either end and near the face plates, a third cutting tool is used for any "middle length" pin or journal on the same axis that may require turning.

A support near the tool along the middle of the shaft is very important for accurate work.

Grinding Pins.—After the journals and pins are turned they

are ground on a lathe in which a similar system of two face plates is used, which again allows for quick adjustments for the various positions of the journals and pins. The grinding wheel has the usual movement towards and away from the shaft and also a lengthwise movement, so that the grinding is a very rapid operation.

The fillets also are ground in the same lathe using a different grinding wheel.

During the grinding the sides of the webs are slightly touched in places by the abrading wheel, but no attempt is made to grind them smooth.

Time and Labor Saved.—These two operations of sawing and turning with special apparatus are so effective that for years this firm has been able to compete successfully with firms that manufacture their own material and likewise make the finished product.

SCALPING METAL.

In an establishment for making phosphor-bronze and other special material of like nature, after the slabs are cast and rolled once the entire surface of the slab is "scalped." This is done to remove the skin with its uncertain composition, rendering the slab unsuitable for use where most pure material must be employed. The operation also allows of an accurate inspection of the surface for possible defects or segregation and the digging out of deep impurities.

The machine is of the shaper order, with the cutting operation done during the pull towards the machine instead of, as usual, during the push from the machine. From the head of the shaper there extends an arm working on a hinge, and on the end of this arm is the holder for the cutting tool. This allows of a vertical motion of the holder and tool. In addition the bed holding the slab being scalped is easily pushed sideways by a quick-action wheel and gear, and raised or lowered as desired by a lever pressed by the foot of the workman.

The cutting tool is of steel of about $\frac{5}{8}$ inch by $1\frac{1}{4}$ inches, and the cutting edge is on the bottom and of the wider dimension. This edge is shaped so that it tends to dig into the metal and scrape or scalp it off. It is like the finger nails scraping the top of a cake of wax. The tool is inclined about 15 degrees from the vertical, the top farthest out from the head of the shaper. This angle assists in giving the proper grip to the cutting edge. A cam lifts the cutting tool on the return stroke and a heavy spring pushes it down when cutting.

The action is rapid. Each completed stroke takes less than half a second and each stroke gouges off a bit of the surface, the cut-off pieces of the metal looking like shavings as they fall. The operator rapidly shifts the position of the bed and pays no particular attention to having one cut follow accurately next to the preceding one. He guides the tool across once and then back and again along, as seems necessary to him, occasionally digging out a deep groove where a deep pit shows. The resulting appearance of the surface is of a lot of irregular gouges on the bright metal. The material is clamped very simply in place and three operations are required, as the stroke of the shaper is only about a foot. Little time is lost on the shifting endwise to complete the three sets of scalps or in doing the same to the other side. The ends and sides are not scalped, but are sheared off.

MACHINE FOR WIRE DRAWING PHOSPHOR-BRONZE.

A machine was noted where there were on each side of it six drawings of wire at the same operation without annealing the wire between the drawings. The wire was on a roll at one end and passed at once through the first die, the power for the drawing being supplied by a disc beyond the die. The disc was power driven, and two turns of the wire allowed enough grip to pull the larger portion through the die, the last three discs having three turns of the wire. From the first disc the wire led to the second die, the power supplied by a disc beyond in the same manner. Thus two sets of six operations

went on within a length of twelve feet and the wire was finally wound automatically on spools. Nobody was near the machine, which was automatic, and required occasionally a new bundle of wire.

Of course each disc had to revolve faster than the preceding one, to allow for the increased length of the wire, the ratio from slowest to fastest being about twenty-four to sixty-four revolutions per minute. The wire was reduced in diameter from .220 to .091 inch and the power required was 100 I.H.P.

METHOD OF TESTING SAFETY VALVES AT THE U. S. NAVAL EXPERIMENT STATION.

By ENSIGN L. R. FORD, U. S. NAVY, MEMBER.

OBJECT OF TESTS.

1. The object of most of the safety-valve tests made at the U. S. Naval Experiment Station is to determine whether the valves purchased for installation on naval vessels fulfill the requirements of the specifications prepared by the Bureau of Steam Engineering. Occasionally special tests are made to try out new types of valves, or in compliance with the request of other Departments of the Government, but the tests of naval valves are characteristic of the test methods at the Station.

2. The requirements for the safety valves of the U. S. S. *Nevada* are characteristic of the requirements for most valves contracted for at the present time for use in the service. They are covered by the following extracts from the machinery specifications:

(*a.*) "The valve, while discharging, must be quiet and free from chattering. Simmering will be allowed at 2 pounds before popping, but the valve closure must be sharp."

(*b.*) "The valve must be adjustable from a 5-pound to a 12-pound blowdown, without disassembling the valve, and without causing chattering."

(*c.*) "The spring will be of material round or square in cross-section and of such length and diameter that the fiber stress will not exceed 48,000 pounds per square inch when the spring is compressed to the position it would assume were the valve lifted so as to give its rated discharge."

(*d.*) "Two boiler safety valves, and such other safety valves as the Bureau of Steam Engineering may direct, shall be

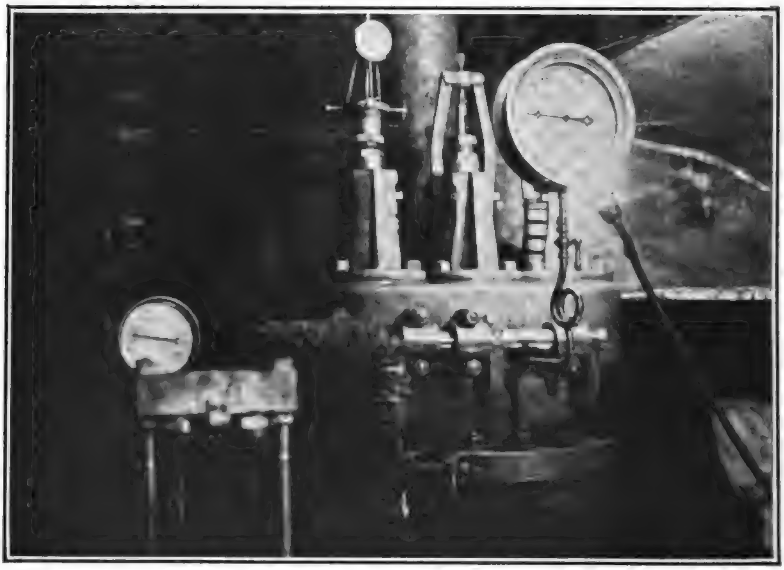
selected at random by the Inspector of Machinery and forwarded to the Engineering Experiment Station, Annapolis, Md., for test. The valves when lifted by steam must lift one-tenth of an inch to fulfill the requirements of the specifications."

3. During the progress of these tests data are obtained that are often of value in preparing new specifications and improving existing designs.

GENERAL ARRANGEMENTS FOR TESTS.

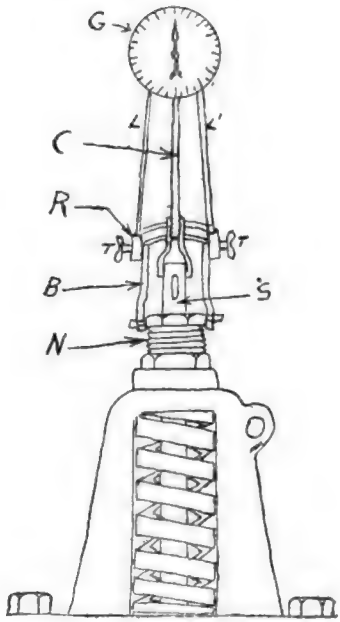
4. For all safety-valve tests the valves are mounted on a Mosher, marine-type, oil-burning boiler containing 4,537 square feet of heating surface. The steam drum is 48 inches in diameter and 14 feet long. A 6-inch gate valve is bolted to the drum nozzle, and the inlet flange of the valve to be tested is usually clamped to this valve. Quick regulation and sure control of the steam pressure is afforded by the oil burners. If increased pressure is desired more burners can be quickly lighted, and if complete cessation of evaporation is desired it is the work of but a few seconds to cut out all the burners. This close control of pressure expedites the work and saves fresh water. The exhaust steam is discharged through a sheet-iron pipe into the atmosphere outside of the boiler compartment. In the case of a small valve this discharge is led to a condenser, but if a large valve is being tested the quantity of steam discharged is too great to be handled by the small condensers at the station.

5. The lift of the valve is measured by a micrometer lift gage, which may be seen mounted on one of the valves in the photograph. The details of this gage and the manner of connecting it are shown in the sketch. A base B is shaped at its lower end to fit over the spring-adjusting nut N, and is secured to this nut by small set screws. The gage G is supported by the standards L and L′, on a bottom ring R. This ring is secured to the base by means of the thumb-screws TT. Thus all parts of the gage except the connect-



Arrangement of Safety Valves and Fittings for Test.

ing rod C are connected rigidly to the body of the safety valve. This connecting rod is secured to the moving stem of the valve. Any lift of the valve is, by means of the connecting rod, communicated to the rack and pinion inside of the gage, and this rack and pinion moves the pointer. The face of the dial is graduated to read to thousandths of an inch.



The steam pressure is indicated by two pressure gages, one mounted directly on the boiler drum, the other connected to the gate valve above its seat. The latter valve can be seen at the left of the photograph. The large gage shown at the right of the photograph is mounted on the safety-valve casing to indicate the pressure in the discharge chamber of the valve.

6. After the tests for lift, simmering, blowdown, chattering and closure are completed the capacity tests are made. The valve is allowed to blow continuously for one, or sometimes two hours. A measure of the weight of steam discharged by the valve is obtained by measuring the water fed to the boiler, since there are no outlets for steam from the boiler, other

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than through the valve, the feed pump being operated by steam from another boiler. The water is measured by the drop of water level in a calibrated feed tank. The water level in the boiler is noted at the same instant as in the feed tank. In this reading of the water level in the boiler lies the greatest source of error in the test. The level is read from a gage glass at the end of the drum farthest removed from the steam nozzle in order to get a minimum fluctuation of level, but it is possible to make an error of $\frac{1}{8}$ inch in the readings. If the errors at beginning and end of run cancel each other there will be no error in the total discharge, but if the errors are additive, it will introduce an error of $\frac{1}{4}$ inch in the drum water level, equivalent to about 65 pounds in the total discharge. If the valve under test is small enough for the discharge to be handled by the condenser an accurate measure of the discharge is obtained by weighing the condensed steam. This cannot be done, however, if the pressure in the discharge chamber of the safety valve is above atmospheric pressure, because there will be a leakage of steam from the casing. When the discharge pressure is below atmospheric pressure the condensed steam and the entering feed are both weighed, one being a check on the other. The quality of the steam discharging through the valve is determined by means of a throttling calorimeter. The top of the attached mercury column can be seen in the attached photograph. The sampling pipe of the calorimeter projects about one-half inch into the steam current. The discharge pipe is run to the bottom of a tank of cold water, mounted on platform scales, so that the amount of water discharged by the calorimeter can be measured. To make the capacity run the number of burners in operation is increased until the valve remains open at the desired lift. This lift is maintained very close to the desired value by regulating one burner under the boiler, by signals to the fireman. When the conditions become constant the run is started, the steam pressure and valve lift being read at one minute intervals, while the feed-tank level, boiler-drum water level, calorimeter pressure and temperature and safety-valve casing

pressure are noted, at five-minute intervals. Popping pressure and blowdown observations are made just before and just after each run.

7. To provide against the introduction into the weight of steam discharged of an error due to leaks around the boiler and its fittings, a leakage test, of the same duration as the capacity test, is made, after the capacity test has been completed. The feed pump is shut down and enough fire is maintained in the furnace to keep the same steam pressure in the boiler as was used in the capacity test. At frequent intervals the level of the water in the boiler is read and at the end of the test these readings are plotted. Since the boiler used is very tight, the curve usually droops off very slightly at the start and then remains a straight line for the rest of the test.

8. A platform of metal grating, built level with the top of the drum, serves as a working platform for the observers. Ventilation is provided by an air duct from the forced-draft blowers, discharging above the level of the working platform. Mounted on the top of the uptake, within easy reach from the operating platform, are switches for signaling to the fireman. Electric lights mounted within view of the fireman are operated by these switches, and indicate to the fireman when the pressure is to be raised or lowered.

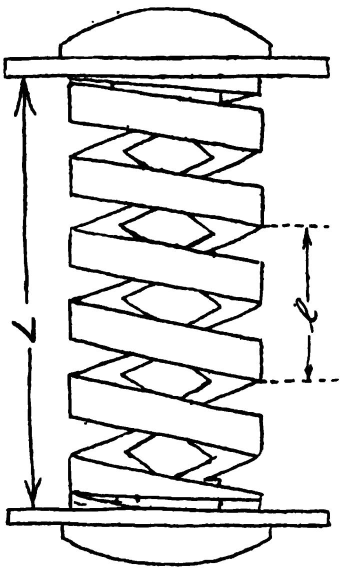
9. All the instruments used for the capacity, lift and blowdown observations are carefully calibrated and calibration sheets made out, to be used in the final calculations.

TEST OF SPRINGS.

10. After the tests for lift and blowdown are made, and before the capacity runs are started, the valve is dismantled for examination and test of the springs.

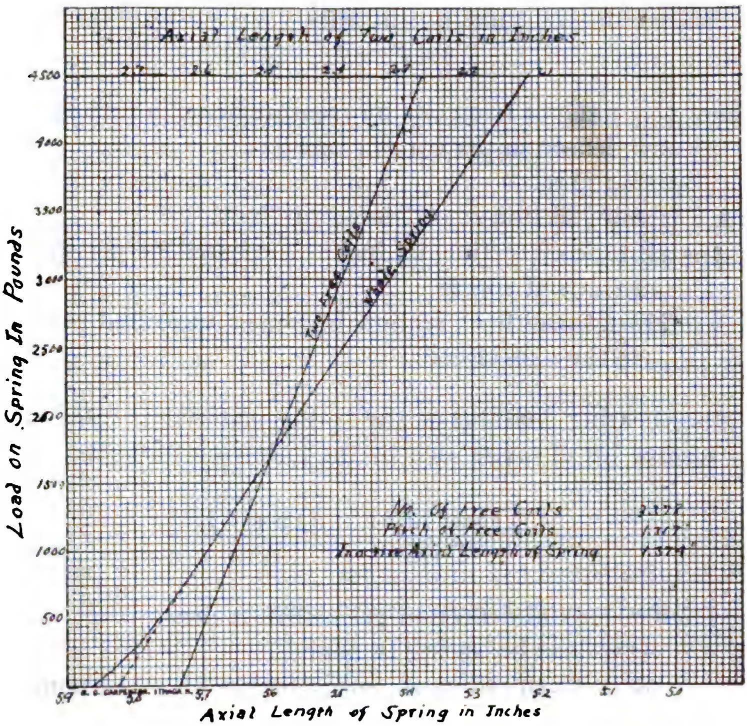
11. The springs are compressed in a testing machine, between two cast-iron discs with spherically-shaped ends, as shown in the sketch. A lug on the face of each disc fits into the spring and serves to hold it into position. The length L is measured by micrometers for various loads, from zero to the

maximum, then to zero again. Due to the formation of the ends of the spring to form a solid bearing, the number of coils available for actively resisting the deflection of the spring is less than the total number of coils. At loads near zero all the coils act, but as soon as the load increases, the tapered coils at the ends press against the next coils and become inactive. This is shown on the attached curve, where the line of deflection curves to the left when the end coils are active. To obtain the number of free coils and the inactive length of



the spring a curve of deflection is plotted, using the length L in the sketch. A curve of deflection for two free coils for the same loads is also plotted, using the length *l*. In the attached curve the deflection of the whole spring, for a load of 4,500 pounds is 5.824 — 5.216 = —0.608 inch, while the deflection of the two free coils is 2.633 — 2.273 = 0.360 inch, or the deflection of one free coil is 0.360 ÷ 2 = —0.180 inch. Then the number of free coils in the spring is 0.608 ÷ 0.180 = 3.378. At loads near zero all the coils become active, as shown by the curvature of the line of deflection. The length

5.824 is obtained by prolonging the straight part of the line. Take any load, say 2,500 pounds, then if the length of two active coils is 2.434 inches the active length of the spring at a load of 2,500 pounds is $3.378 \times 2.434 \div 2 = 4.111$ inches. Since the total length of the spring at this load is 5.484 inches, the inactive length is $5.484 - 4.111 = 1.373$ inches.



12. The maximum fiber stress in the spring occurs at the middle of the inside edge of the section of the spring coil and is made up of the four component stresses due to torsion, direct shear, bending and direct compression. The effect of the last two is slight enough to be neglected. The first two produce a maximum shearing fiber stress, which is calculated for a valve lift of 0.100 of an inch by the following formula:

$$S = \frac{9\,Q\cos\theta\,R}{2bh^2} + \frac{Q\cos\theta}{A}$$

in which Q = load on spring in pounds.

R = mean radius of coil measured from the axis of spring to center of gravity of section in inches.

θ = angle of inclination of coil to plane perpendicular to axis of coil.

$$\cos\theta = \frac{2\pi R}{\sqrt{(p-d)^2 + (2\pi R)^2}}$$

where p = pitch of free coils by measurement, in inches.

d = deflection per coil, in inches.

$\pi = 3.1416$.

A = area of section of coil in square inch.

h = altitude of section in inches measured perpendicular to axis of spring.

b = Mean breadth of section in inches measured parallel to the axis of coil.

The shearing modulus of the spring material is calculated for a valve lift of 0.100 inch by the following formula:

$$d = \frac{Q\cos^2\theta\, R^2\, L}{C\, J\, G} + \frac{Q\cos^2\theta\, L}{A\, G},$$

in which d = axial deflection per coil in inches.

Q = load on spring in pounds.

R = Mean radius of coil in inches by measurement.

p = pitch of coils in inches.

$L = \sqrt{(p-d)^2 + (2\pi R)^2}$ = length of one free coil in inches actively opposing the compression of the spring.

$$\cos\theta = \frac{2\pi R}{L}.$$

C = St. Venants constant for the resistance to torsion of bars of nearly square section.

$A = h \times \frac{B + b}{2}$ area of section in square inches.

h = altitude of trapezoidal section in inches.
B = larger base in inches.
b = smaller base in inches.

$$J = \frac{h}{48}\left[B^3 + B^2b + Bb^2 + b^3\right] + \frac{h^3}{36}\left[\frac{B^2+4Bb+b^2}{B+b}\right]$$

= polar moment of inertia of trapezoidal section in inches.

G = Shearing modulus of elasticity in pounds per square inch.

The numerical value of the shearing modulus of elasticity is obtained by substitution in this formnla.

13. After the tests are completed the valve is set for the proper popping pressure on the boiler and is shipped either to the manufacturer or to the ship on which it is to be used.

14. If the valve develops any defect during the tests, or fails to function properly, the manufacturers are notified and given the opportunity to send a representative to make such changes or adjustments as he may consider desirable.

15. At the conclusion of the tests the engineer in charge of the test makes a report to the Head of the Experiment Station, containing in detail a description of the conduct of the test, a recommendation that the valve be accepted or rejected, and such other remarks or recommendations as he may consider pertinent or desirable. This report usually contains information and data derived and tabulated as follows:

1. Dimensional data.
   From measurements.
2. Popping pressure under valve, pounds per square inch.
   From observation corrected for calibration error.
3. Popping pressure in boiler drum, pounds ÷ square inch.
   From observation corrected for calibration error.
4. Average popping pressure, pounds per square inch gage.
   Average values of items 2 and 3.

5. Average popping pressure, pounds per square inch absolute.
   Item 8 + 14.7 pounds.
6. Closing pressure under valve, pounds per square inch gage.
   From observation corrected for calibration error.
7. Closing pressure in boiler drum, pounds ÷ inch gage.
   From observation corrected for calibration error.
8. Average closing pressure, pounds ÷ square inch gage.
   Average of items 6 and 7.
9. Average blow down, pounds ÷ square inch.
   Item 5 — item 8.
10. Average maximum lift, inches.
    From observation corrected for zero setting error.
11. Average lift at closing, inches.
    From observation corrected for zero setting error.
12. Area of opening at maximum lift, inches.
    From formula

$$A = \pi DL \frac{2\sqrt{R^2 - \frac{D^2}{4}} - L}{2\sqrt{R^2 + L^2 - 2L\sqrt{R^2 - \frac{D^2}{4}}}},$$

in which L = lift of valve in inches, item 14.
D = contact diameter between feather and seat.
R = radius of balled seat in inches.
A = opening area in square inches.

13. Theoretical discharge in pounds per hour.
    From formula

$$W = 3{,}600 \frac{AP}{70},$$

in which W = discharge of dry steam in pounds per hour.
A = area of opening in square inches, item 12.
P = absolute steam pressure in pounds per square inch, item 5.

14. Correction factor from capacity runs.
    Value assumed from measurement of discharge.
15. Probable actual discharge, pounds per hour, item 13 × item 14.

Directly before and after the capacity runs the above items are again observed and tabulated. During the capacity runs the following additional data is observed:

16. Valve casing pressure.
17. Calorimeter pressure, inches of mercury.
    From observation.
18. Calorimeter temperature, degrees F.
    From observation.
19. Steam quality.
    From formula

$$x = \frac{h + k\,(t_s - t) - q}{r},$$

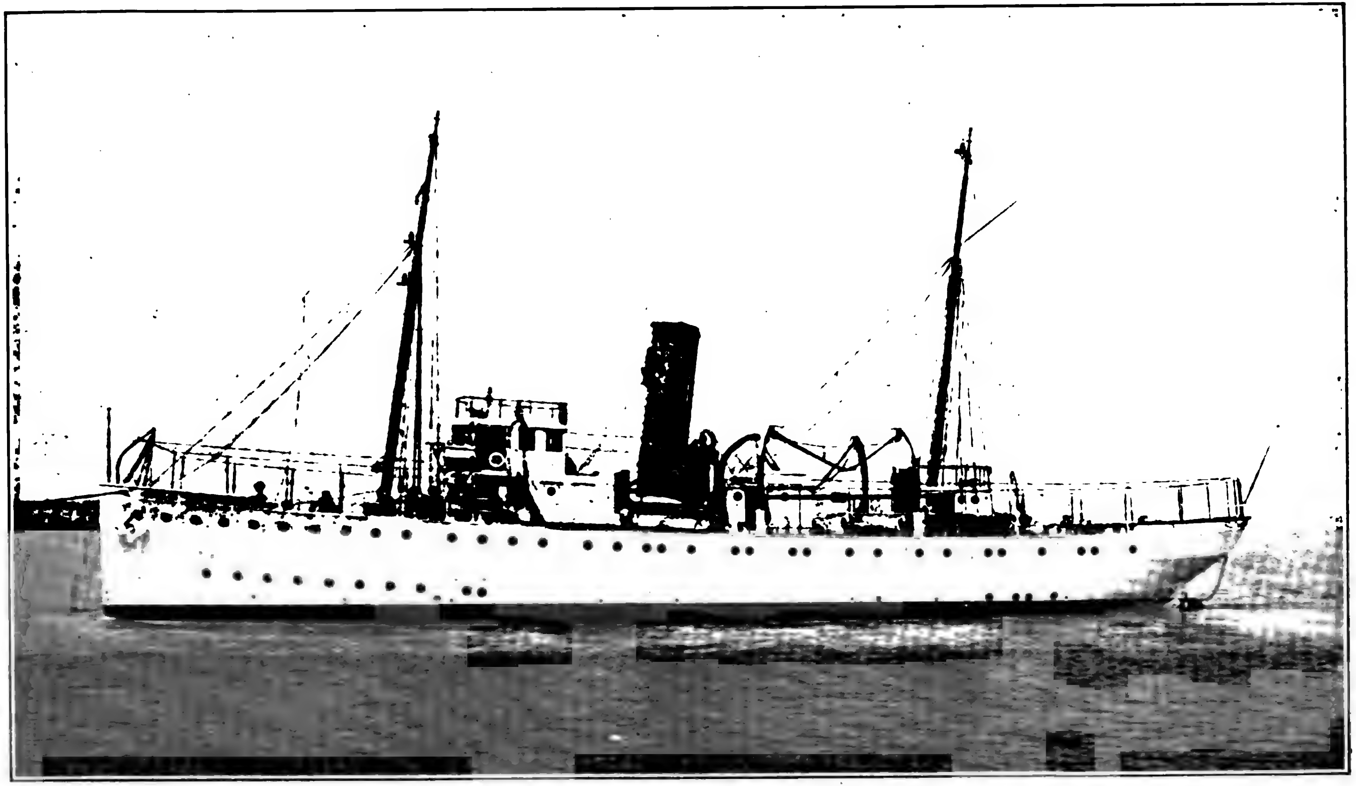
in which x = dryness fraction.
r = latent heat, and
q = heat of the liquid at the blowing pressure.
h = total heat.
t = temperature corresponding to the pressure in the calorimeter.
t_s = temperature in the calorimeter.
k = proper value of the specific heat of the steam in the calorimeter.

20. Lift of valve, inches.
    From observation.
21. Water fed to boiler, pounds.

From observation of change of water level in feed tank, corrected for change of water level in boiler drum and for calorimeter discharge.

22. Steam discharged by safety valve, pounds per hour.
    Item 21 × item 19, divided by number of hours of test.
23. Correction factor.
    Item 22 ÷ item 13.

16. In addition to the above data the report includes the result of the spring tests, with the calculations as previously explained, and with curves of deflection, of which the attached curve is characteristic.



Coast Guard Cutter "Ossipee."

TRIALS OF COAST GUARD CUTTERS *OSSIPEE* AND *TALLAPOOSA*.

By SECOND LIEUTENANT OF ENGINEERS W. M. PRALL, U. S. C. G., MEMBER.

An Act of Congress, approved June 24, 1914, authorized the construction of two Coast Guard cutters, one for duty on the coast of Maine, replacing the cutter *Woodbury*, and one for duty in the Gulf of Mexico, replacing the cutter *Winona*, these two vessels being old and no longer fit for service.

In determining the size, type and power of the new cutters, Coast Guard Headquarters, after carefully considering the work which these ships would be required to perform, and the available appropriation, designed single-screw steel vessels somewhat smaller than the average sea-going cutter, having full lines, moderate power, and sufficient capacity for fuel and stores to insure keeping the seas for long periods, this latter quality being of great importance when searching for derelicts. Seaworthiness, reliability and ability to render efficient aid to vessels in distress, regardless of weather conditions, received first consideration; speed being a matter of secondary importance.

On September 15, 1914, contracts for building the ships were awarded to the Newport News Shipbuilding and Dry Dock Company of Newport News, Virginia, at their bid of $198,000 each. Keels were laid November 17, 1914, and the vessels launched on May 1, 1915. During construction the ships were designated as Coast Guard Cutters Nos. 26 and 27. No. 26, for duty on the Maine coast, was christened *Ossipee*; No. 27, for duty in the Gulf of Mexico, was christened *Tallapoosa*.

Standardization and full-power speed trials of the *Ossipee* were run on June 22, 1915; trials of the *Tallapoosa* were run on June 29, 1915, both trials being successful in every respect.

On July 8th and 15th, 1915, respectively, the ships were accepted by the Government.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	165–10
on L.W.L., feet and inches	150–00
Breadth, moulded, feet and inches	32–00
Depth, moulded, at side, feet and inches	20–09
Draught to L.W.L., feet and inches	11–06
Displacement at above draught, tons	908
Tons per inch immersion at L.W.L.	8.4
Block coefficient	.58

The two vessels are built to the same lines except that the forward underwater body of the *Ossipee* is modified by cutting away the forefoot, this being done to enable the vessel to better cope with the winter ice conditions which will be encountered on the Maine coast. This vessel is equipped for burning coal, but, because of the low price and facility with which oil fuel can be obtained at ports on the Gulf coast, it was deemed advantageous to fit the *Tallapoosa* as an oil burner. On this vessel is also installed an Audiffren-Singrun refrigerating apparatus, which is omitted on the *Ossipee*. These features constitute the only differences in the two ships.

The vessels are of the flush-deck type, with berth deck forward and aft. There are seven watertight bulkheads, two pole masts, schooner rigged, and one funnel.

DESCRIPTION OF MACHINERY.

The main engine is of the vertical, inverted, three-cylinder, triple-expansion type, arranged with high-pressure cylinder forward and low-pressure cylinder aft. All main valves are of the piston type worked by Stephenson link motions.

The main condenser is cylindrical with cooling surface of 1,328 square feet. The main circulating pump is independent and of the centrifugal type. The main air pump and two bilge pumps are attached and operated from the crosshead of the low-pressure cylinder.

The principal details of machinery are as follows:

Main engine, H.P. cylinder, diameter, inches	17
valve, diameter, inches	8
M.P. cylinder, diameter, inches	27
valve, diameter, inches	14
L.P. cylinder, diameter, inches	44
valves (two), diameter, inches	14
stroke of pistons, inches	30
travel of valves, inches	5½
diameter of main steam pipe, inches	6
exhaust pipe at engine, inches	15
condenser, inches	16
Crank shaft, built-up, length overall, feet and inches	14–7¾
diameter, main journals, inches	8¾
crank pins (two-inch hole), inches	8¾
length, first five main journals, inches	10
after main journal, inches	12
crank pins, inches	9
Thrust shaft, length overall, feet and inches	10–1½
diameter, inches	8¾
number of thrust collars, inches	8
diameter of collars, inches	17
thickness of collars, inches	1¾
length of steady bearings (two), inches	9
Intermediate shaft, length over-all, feet and inches	12–8
diameter, inches	8½
length of spring bearing, inches	11
Propeller shaft, length overall, feet and inches	15–2
diameter, inches	9½
length of forward bearing, feet and inches	2–0
after bearing, feet and inches	3–0
thickness of composition casing, forward bearing, inch	0 13/16
thickness of composition casing, after bearing, inch	0¾
composition casing, after bearing, inch	0½
All shaft couplings, diameter of discs, inches	18
thickness of discs, inches	2½
Boilers, Babcock & Wilcox, marine type, water tube, number	2
length of generating tubes (exposed), feet and inches	9–0
diameter of generating tubes (lower row), inches	4
(all others), inches	2
height over casing, feet and inches	10–2¼
width over casing, feet and inches	11–4½
diameter of drum, inches	42
heating surface (each boiler), square feet	2,250
grate surface (each boiler), *Ossipee*, square feet	62

Smoke pipe, common to both boilers, diameter inside, feet and inches.	4–11
diameter outer casing, feet and inches	5–9½
height above grates, feet and inches	47–0
Propeller, built-up, manganese bronze, four blade, R. H., diameter, feet and inches	9–6
pitch, feet and inches	12–0
developed area, square feet	39.1
Condenser, cylindrical, cast-iron shell, diameter inside, feet and ins.	3–6
length between tube sheets, feet and inches	7–3⅝
diameter of tubes (No. 18 U. S. S. G.), inch	0⅝
number of tubes	1,112
cooling surface, square feet	1,328
Circulating pump, centrifugal, diameter of suction, inches	8
diameter of impeller, inches	32
engine cylinder, inches	5
stroke of engine piston, inches	6
Feed-water heater, Griscom-Russell multicoil, diameter of shell, ins.	16½
length of shell, feet and inches	5–4½
number of coils	8
Evaporator, Griscom-Russell multicoil, diameter of shell, inches	30
length of shell, feet and inches	6–6
number of coils	8
capacity, gallons per 24 hours	2,300
Distiller, Griscom-Russell multicoil, diameter of shell, inches	10
length of shell, feet and inches	4–3
number of coils	2
Steam steering engine, manufactured by Hyde Windlass Company:	
Diameter of cylinders (two), inches	5
Stroke of pistons, inches	5½
Windlass engine, manufactured by Hyde Windlass Company:	
Diameter of cylinders (two), inches	6
Stroke of pistons, inches	8
Gypsy engine (located aft) manufactured by Hyde Windlass Company:	
Diameter of cylinders (two), inches	5
Stroke of pistons, inches	8

The electrical installation consists of twin sets of Westinghouse direct-current, compound-wound generators, turbine driven through reduction gear.

Capacity (each generating set), kw	15
Voltage	110
R.p.m., turbine	7,200
R.p.m., generator	1,800

The radio apparatus, manufactured by the Marconi Wireless Telegraph Company of America, consists of a main set of

2-kw. capacity and an auxiliary set of one-half-kw. capacity, which can be operated either by storage battery or the ship's main generators. Each vessel also carries a ½-kw. portable radio set made by the Wireless Specialty Apparatus Company. The portable sets are operated by hand-power generators.

FUEL AND WATER CAPACITIES.

Coal bunker capacity, *Ossipee*, tons, ...	213
Fuel-oil capacity, *Tallapoosa*, gallons, ...	53,200
Fresh-water capacity (for all purposes), *Ossipee*, gals..	11,845
Fresh-water capacity (for all purposes), *Tallapoosa*, gallons, ...	13,345

RECIPROCATING PUMPS.

Number.	Make and Type.	Purpose.	Diameter steam cylinders, inches.	Diameter water or air cyls., inches.	Stroke, inches.
1	N. N. S. & D. D. Co., vertical, single acting, attached.	Main air	...	20	11
1	Blake & Knowles, vertical, simplex, double acting.	Main feed	10	6	12
1	Blake & Knowles, vertical, simplex, double acting.	Fire & Wrecking. also aux. feed.	18	10	18
2	N. N. S. & D. D. Co., vertical, single acting, attached.	Bilge	...	3½	11
1	Blake & Knowles, horizontal, duplex.	General service	7½	6	10
1	Blake & Knowles, vertical, simplex.	Fresh water	3½	4	4
1	N. N. S. & D. D. Co., double acting.	Hand deck	...	5	...
1	Warren, horizontal, duplex.	Oil fuel supply *Tallapoosa* only.	5½	6	6
2	Warren, horizontal, duplex.	Oil fuel service *Tallapoosa* only.	4½	2¾	4

TRIALS.

As engines practically identical with those described above have been installed and proved satisfactory in cutters previously built, and as the qualities of Babcock & Wilcox

STANDARDIZATION TRIALS, U. S. COAST GUARD CUTTERS.

Number of run.	*Ossipee*, July 22, 1915.						*Tallapoosa*, July 29, 1915.					
	Speed, knots.	R.P.M.	I.H.P.	Means.			Speed, knots.	R.P.M.	I.H.P.	Means.		
				Kts.	R.P.M.	I.H.P.				Kts.	R.P.M.	I.H.P.
1	7.63	55.29	77.2				6.27	51.32	68.4			
2	4.58	55.89	77.0	6.27	56.55	78.8	4.65	52.37	79.7	5.63	52.39	76.5
3	8.29	59.13	84.2				6.96	53.57	78.2			
4	7.35	82.77	217.1				7.45	82.12	240			
5	10.52	84.95	240.4	9.03	84.51	237.4	10.12	83.15	256	8.80	83.02	250.8
6	7.72	85.38	251.8				7.50	83.65	251			
7	11.69	98.42	407.9				8.91	97.14	397			
8	8.90	97.54	398.0	10.29	98.25	403.0	11.53	97.03	405	10.28	97.40	405.3
9	11.66	99.49	....				9.16	98.41	414			
10	9.98	107.96	569.0				12.37	108.30	568			
11	12.15	108.70	604.0	11.12	108.41	590.0	10.08	108.98	593	11.23	108.93	581.3
12	10.21	108.27	583.0				12.37	109.47	571			
13	11.80	130.0	1,049				11.73	127.9	954			
14	12.87	130.67	....	12.38	129.81	1,050	13.00	129.59	1,037	12.47	129.65	1,024.3
15	11.98	127.90	1,051				12.13	131.6	1,063			

U. S. Coast Guard Cutters.

TWO-HOUR FULL-POWER TRIALS UNDERWAY.

(Run in Hampton Roads and Chesapeake Bay.)

	Ossipee.	*Tallapoosa.*
Date of trial	6/22/15	6/29/15
Mean draught, feet and inches	10–4½	9–11½
Displacement, tons	785	748
Boilers in use	2	2
Heating surface	4,500	4,500
Grate surface	124	......
Draft, natural	.18	.21
R.p.m.	133.9	134.0
Pressures:		
Boiler, gage	180	172
H.P. chest	169	164
First receiver	64.8	60.0
Second receiver	11.2	9.0
Vacuum	27.5	27.8
M.E.P. — H.P. cylinder	76.2	76.7
M.E.P. — M.P. cyinder	36.2	35.9
M.E.P. — L. P. cylinder	13.9	11.8
M.E.P. referred to L.P	39.0	36.8
I.H.P. — H.P. cylinder	349.4	351.7
I.H.P. — M.P. cylinder	419.2	415.5
I.H.P. — L.P. cylinder	426.4	363.8
I.H.P. — total	1,195.0	1,131.0
Speed in knots	12.55	12.66
Temperatures:		
Outside air	96	86
Working platform, E.R	99	99
Fireroom	129	107
Stack	521	430
Main injection water	73	74
Outboard delivery	105	109
Main feed	186	184
Flue gas analysis:		
Per cent. CO_2	13.5	14 3
Per cent, O	3.7	2.2
Per cent. CO	0.7	0.08
Fuel	New River coal.	Texas oil.
B.t.u	......	18,792
Flash (degrees F.)	......	221
Specific gravity	......	0.907

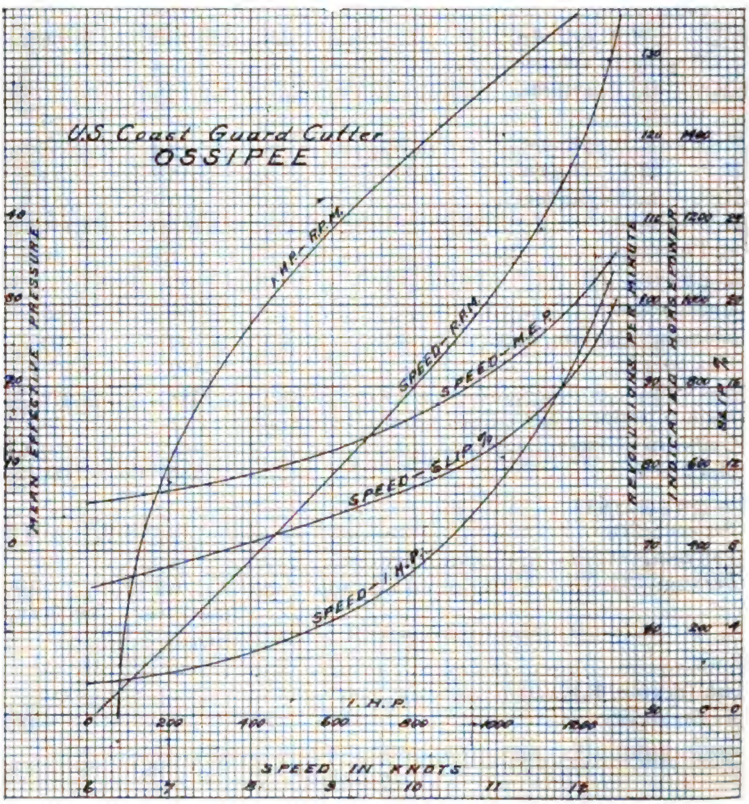
boilers have been well tested by actual service on vessels of the Coast Guard, the only trial requirements were as follows:

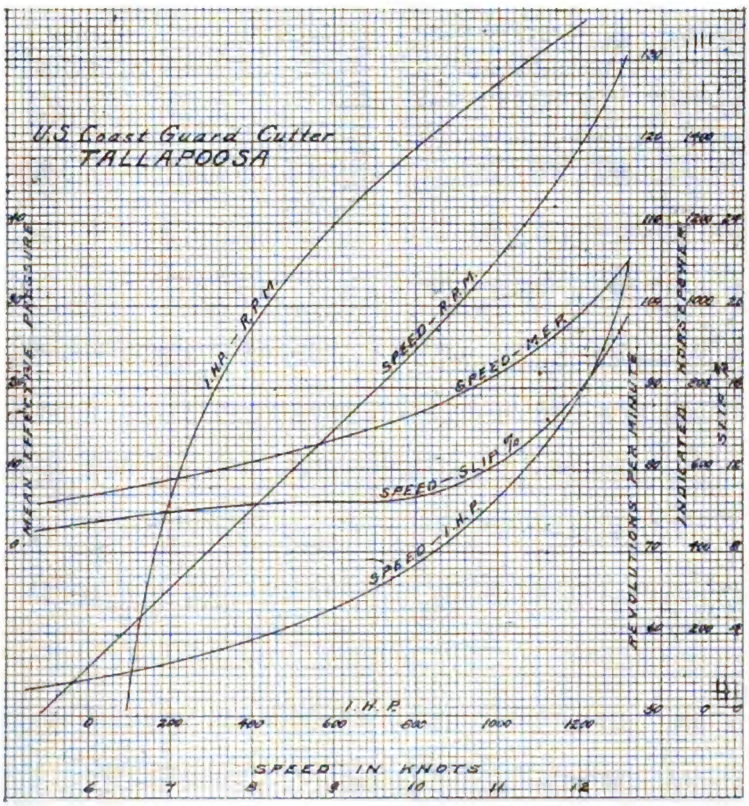
(*a*) An official dock trial consisting of the operation of the machinery at the dock until it could be run at full power for four consecutive hours without heating of bearings or necessity for adjustments.

(*b*) Standardization trials, run between six knots and full speed to determine the relation between revolutions, indicated horsepower and speed.

(*c*) A full-power trial, consisting of a run underway for two consecutive hours, working off at 180 pounds pressure all the steam which the boilers would generate under natural draft.

Results of the underway trials are shown in the accompanying curves and tables.




U.S. Coast Guard Cutter
TALLAPOOSA
I.H.P. – R.P.M.
SPEED – R.P.M.
SPEED – M.E.P.
SPEED – SLIP %
SPEED – I.H.P.
MEAN EFFECTIVE PRESSURE
REVOLUTIONS PER MINUTE
INDICATED HORSEPOWER
SLIP
I.H.P.
0
200
400
600
800
1000
1200
SPEED IN KNOTS
6
7
8
9
10
11
12

TALLAPOOS .

FULL POWER TRIAL, JUNE 23 '15.

CARD #2	H.P.	I.P.	L.P.
M.E.P.	60	37.6	12
CUT-OFF	.722	.718	.715
I.H.P. PER CYL.	362	435	370
SPRING	100	40	16
TOTAL I.H.P.			1167
R.P.M.			134½
M.E.P. TO L.P.			37.6
BOILER PRESSURE			173
1ST REC.	63	2ND REC.	10
VACUUM			27.8

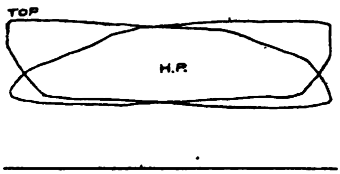
FULL GEAR.

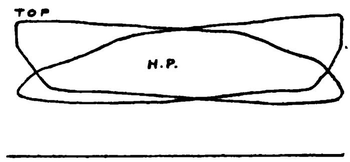
OSSIPEE.

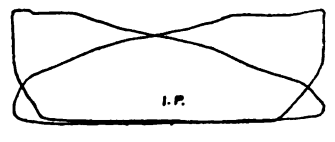
FULL POWER TRIAL, JUNE 22 '15.

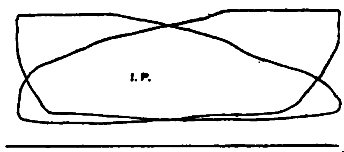
CARD # 4	H.P.	I.P.	L.P.
M.E.P.	72.5	38.8	14.8
CUT-OFF	.722	.718	.715
I.H.P. PER CYL.	329	450	459
SPRING	100	40	16
TOTAL I.H.P.			1238
R.P.M.			135
M.E.P. TO L.P.			33.75
BOILER PRESSURE			180
1ST REC.	71	2ND REC.	13
VACUUM			27.5

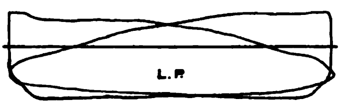
FULL GEAR.

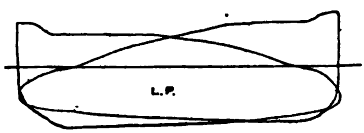












ELECTRIC-ARC WELDING.

By LIEUTENANT C. S. McDOWELL, U. S. NAVY, MEMBER.

Welding is the joining of two pieces of metal of like or unlike characteristics by fusion, while in the plastic state. The old definition of welding was the process of uniting two pieces of metal by hammering them together while hot enough to be plastic. Modern methods, however, of obtaining high temperatures by means of gases and electricity, has broadened the definition of welding and brought in use additional processes, to which the term "welding" has been applied. It is the purpose in this article to describe only the Electric-Arc Welding process, which it is predicted will rapidly become the standard method of joining sheet metals of all thicknesses, reclaiming castings, repairing broken machinery of all kinds, building up of worn parts, welding seams in new boilers, tanks, etc., making of high-speed tools, repairing boilers, etc., and an arc-welding equipment will be a necessary adjunct to every properly-equipped machine shop.

Conditions for Successful Welding.—The essential characteristics of a successful weld are: That the metal in the welded joint shall be free from impurities, slag and defects of all sorts; that it shall possess a sufficient amount of elongation, flexibility and tensile strength; and that the process of welding shall be such as to reduce to a minimum disturbances in the texture of the surrounding metal. In certain classes of work flexibility and elongation in the weld is of more importance than tensile strength.

The quality of the weld obtained with electric welding is dependent on the folowing:

1st. The furnishing of the correct amount of energy at the weld for obtaining of the proper working temperatures of the material to be welded.

2d. The quality of the metal electrodes (the welding wire).

3d. The skill of the operator.

The Correct Amount of Energy.—The material worked on, the anode, is heated by the impinging of the cathode stream, by the julean heat developed by the ohmic contact resistance and by the radiant heat given off by the arc; the metallic pencil or cathode is heated by the julean heat due to its ohmic resistance to current flowing and by the slight contact resistance at the electrode surface. The greater proportion of the heat generated at the arc is at the anode, or material being welded (approximately 75 per cent. of the total heat), when the metallic arc is used. This is quite satisfactory due to the greater means of dissipation at the anode by radiation, convection and conduction. Practically all the energy supplied to the arc is dissipated as heat, only a slight proportion being given off as light, and to regulate the amount of heat units and thereby the temperature of a particular shape and class of material, it is essential to regulate the amount of energy supplied to the arc as measured in watts. The maximum temperature of steel in the usual converter is approximately 1,800 degrees C., the melting point being approximately 1,400 degrees C., the temperature of boiling steel at atmospheric pressure is approximately 2,450 degrees C., while the temperature of the arc stream may greatly exceed this. The result is that in electric-arc welding steel is being worked near a critical temperature. The material to be welded should be in a plastic state sufficient for the proper intermixing of the metal and obtaining of perfect fusion; if the temperature is too low the added material will not adhere to the original metal and the weld will fall apart at the surface; on the other hand, if too high a temperature is obtained the metal will be burnt and the weld will be greatly weakened by slag thus formed and will be of coarse and irregular structures. The surfaces of the metals to be welded must tend to cohere to a marked extent and the working temperature must be that at which the foregoing condition is most prominent. The best welding condition for iron and steel exists within a limited range

of temperature only. The safe working temperature depends somewhat on the material to be welded; an operator quickly obtains the necessary experience to tell if he has the proper temperature. The amount of energy necessary to obtain the proper temperature at the weld depends upon the size and shape of the piece worked on, it being the amount of energy necessary to supply the heat losses and keep the weld constantly at the proper welding temperature, the amount of energy required varies as the whole mass of the article becomes heated, a greater amount being required at first when the mass is cold; for this reason it is important in order to obtain consistent results to have a control of the energy at the operator's end of the line. This control should require very few changes in the supplied energy and is not to overcome the variation of energy due to changes in arc length. The temperature at the weld should remain practically constant; a momentary inrush of current will burn the metal at that point and cause a flaw with the chances of reducing the tensile strength of the weld 50 per cent. or even more.

Electrodes.—There are two methods of electric-arc welding: one, the Benardes process, in which a carbon electrode is used; and the other, the Slavinoff process, in which the metallic electrode is used. As a result of the tests which have been conducted, and from the experience of others in electric-arc welding, it is believed that the carbon-electrode process is not suited for general work, some of the reasons being that much greater difficulty is experienced in maintaining the proper temperature, and there are more chances of getting an excess of carbon in the weld. In the Slavinoff process, which is nearly universally used at present, it is necessary to have the metal electrodes of such material that the deposited metal in the weld shall have practically the same characteristics as the rest of the metal of the object worked on. As certain of the constituents of the electrode are partially lost in the arc, it is usually necessary to have the electrode contain an excess of certain materials over what is desired in the finished weld. The amount of the loss of these constituents depends upon the

temperature, and it is necessary in order to obtain desired and consistent characteristics in the finished weld to have a constant temperature at the weld. The steel companies will guarantee the results with the electrodes which they supply only if the system of arc welding with which they are used can maintain a constant temperature at the weld.

The Operator.—A certain amount of skill and experience is required of the operator, no matter what system of electric welding may be used; but some types of outfits require much more skill and closer attention than others, and it is considered essential that the ideal system should require a minimum of experience and only normal mechanical skill. A system which depends primarily on the skill of the operator cannot turn out consistent work and is not suited to the Navy's use.

Fluxes.—Certain companies claim that a flux is necessary to obtain good results, but in the tests conducted all sorts of material and in all positions have been welded, and the best results have been obtained from systems in which no flux is used. The claims in favor of the flux are: that it blankets the weld by forming a gas around the material which prevents oxygen reaching it and thereby prevents oxidation; this has been proven not necessary, by making similar welds first where oxygen was entirely excluded, and then under normal conditions in the air, and there was no difference in strength or structure of the weld. Another claim is that the flux acts as a scavenger to remove impurities from the weld; it cannot act in this way unless the metal actually boils, and this is a condition which, as previously shown, should be avoided. There are also certain users who believe a flux necessary for overhead work; but, in tests conducted, as good and consistent welds were obtained when welding overhead without a flux as in any other position. It is considered that in a good electrical welding system a flux is not necessary, and is simply an added expense and complication.

Automatic Control.—While it is recognized that it is desirable to have as simple an equipment as possible, it is considered necessary to have an automatic control of the input

energy to the weld, the reasons for which have been previously mentioned, so that when the proper amount of energy has been determined for a particular job it will remain constant regardless of the varying of the arc length. A system with fixed resistances depends entirely on the skill of the operator in maintaining his arc length constant and thereby the energy constant. This system gives good results at times, but our tests showed that even with a skilled operator, furnished by the manufacturer, tensile strengths varying as much as 50 per cent. on the same class of material were obtained. It should be possible for the operator to set the current controller at the desired amount as well as at the panel board; the controller should automatically keep the current approximately at the fixed value. A variation of less than 5 per cent. can be obtained with a well designed equipment.

Cutting.—The electric arc has been found suitable for cutting, but a carbon electrode must be used; no automatic-current control is necessary, although a choke coil is advisable to prevent large inrushes of current. The amount of current varies with the size of the material to be cut; from 250 to 400 ampères are required for burning off rivet heads and light section plates, while from 500 to 800 ampères may be required on plates 4 inches thick. This is a momentary load, however, and a 300-ampère continuous-duty machine is considered sufficient. It is necessary to cut away the edges of the cut and remove the burnt metal.

Preparation of Material to be Welded.—The material to be welded should be cleaned with a scraper or wire brush to remove oxides and prevent forming of slag, and it is also necessary to bevel the edge sufficiently so that the distance from the electrode to bottom of the weld is less than that of the electrode to any other part of the article, so that the arc will not stray. In thick plates, where possible, and especially in castings, it is usual to weld from both sides, and in this case the original material is pointed by beveling on both sides.

Application of Arc Welding.—During the past year the New

York Navy Yard has had contract electric welding done on boilers of various ships. Certain defective castings have been welded, blow holes filled in others, and miscellaneous repair work has been done while the various machines were under test. Additional uses are being developed as the advantages of this method became better known. A large saving in cost over other methods of repair have been made on boiler jobs, in addition to a saving in time, notwithstanding the large profit which the outside contractors have made on the jobs. A specific application of arc welding is in the making of high-speed tools, a piece of the tool being made of ordinary steel and high-speed tool steel is used for the cutting edge only. This is shown in detail later in this paper.

Some of the various applications are as follows:

Building up of worn wearing parts, pins, rollers, bearings, etc.
Welding of plates in lieu of riveting, or where seams are leaking.
Building up of rivets.
Building up stripped gears.
Repair of cracked castings.
Making of high-speed tools.
Filling blow holes.

In manufacturing work:

Welding of heads on tanks.
Welding of tubes in tube sheets.
Welding of feet and end frames, etc.

Brass, bronzes, and aluminum as well as steel, cast steel, wrought iron and cast iron can be welded, but none of the demonstrators have been able as yet to get very high tensile strength on naval brass.

Special Application for the Navy.—Electric-arc welding is considered especially applicable for use on shipboard for emergency repairs of all sorts. In this connection it may be noted that the British cruiser *Glasgow* put into Rio de Janeiro

after the battle off Chile with several holes below or near the water line, and was able, with an arc-welding set which happened to be in Rio de Janiero, to weld plates over the holes inside of 24 hours and put to sea, taking with them the arc-welding set.

Specific Cases Showing Costs and Savings.—Accurate cost and time data on some specific cases of a widely varying nature have been obtained from one of the large railroad companies. Photographs of the article before and after welding are given, with the times and cost to make the repairs, as compared to other methods.

Figure 1 shows electric welding being subjected to the following treatment:

A piece of boiler plate 18 by 20 inches was cut through the center and welded by the electric-welding process. This sheet was then used as a face and a tank built on the back side, as illustrated in the photograph.

This tank was filled with cold water and fuel-oil burner turned on the center of the weld in the boiler plate. The heat was so intense that the water was made to boil in an average of from 4 to 5 minutes.

This boiling process was continued for 10 minutes, then the water was allowed to cool down. When the water had become cool the burner was again put on sheet and water brought up to boiling point as before.

This operation was repeated nine times, the tenth time the boiling water was poured out and tank immediately filled with cold water and again raised to boiling point. This operation was repeated three times, after which the boiler plate was cut loose from the tank and test strips cut in the regular way.

The average tensile strength of nine strips from this plate was 55,000 pounds per square inch of welding metal.

Figures 2, 3 and 4 show cylinder on engine, class K–4, repaired by electric-welding process, thus saving the application of new cylinder.

The following statement shows cost to repair by above process as against applying new:

Cost of Present Method.

Labor:			
Removing old and applying new cylinder complete			$125.00
Machining cylinder complete			17.50
Cylinder bushing machined and applied			5.85
Valve-chamber bushing, machined and applied			5.40
Turning and fitting bolts and studs			7.00
Cost of transportation			9.00
Cost of loading and unloading and handling cylinder			12.00
			181.75
Material:			
Cylinder, weight 8,550 lbs., at 3 cents per lb.	$256.50		
Hunt-Spiller bushings	60.50		
Bolt and stud material	8.00		
Deduct for scrap value of old cylinder	42.00	$283.00	
Total net cost new cylinder			$464.75
Cost by Electric Welding:			
Labor:			
Preparing cylinder for welding	55.00		
Welding, 11½ hours, at 27½ cents per hour	4.40		
		59.40	
Material:			
14 pounds of welding wire, at 5 cents per lb.	.70		
Electricity, 11½ hours, at 12 cents per hour	1.38		
Bolt and stud material	2.00	4.08	63.48
Total net saving			401.27

Figures 5 and 6 show broken walking beam for air compressor repaired by electric welding for U. S. S. *Drayton*, Brooklyn Navy Yard.

Cost of New Walking Beam:			
Labor:			
Machining complete		$8.10	
Material:			
New casting	$12.00		
Deduct for scrap value of old castings	.60		
		11.40	
Net cost of new walking beam			$19.50
Cost to Electric Weld:			
Labor:			
Cutting out and preparing for weld, 3 hours, at 30 cents per hour	$0.90		
Welding, 2 hours, at 27½ cents per hour	.55		
Drilling, reaming and facing holes	.45		
		1.90	
Material:			
2¼ pounds welding wire, at 5 cents per lb.	.11		
Electricity, 2 hours, at 12 cents per hour	.24		
		.35	2.25
Net saving by electric welding			17.25

Figures 7 and 8 show lathe gears reclaimed by electric welding process.

Statement showing comparative cost to reclaim by above method as against purchasing new gears:

Cost of new 13-inch gears, shaft and pinion		$9.00	
Cost of new 11-inch gear		4.00	
Scrap value of old gears		.20	
Net cost new			12.80
Cost to Reclaim by Electric Welding:			
Labor:			
Welding, 5 hours, at 27½ cents per hour	$1.37		
Cutting teeth	2.80		
		4.17	
Material:			
3½ pounds of welding wire, at 5 cents per lb.	.17		
Electricity, 5 hours, at 12 cents per hour	.60	.77	4.94
Net saving by electric welding			7.86

Figure 9 shows broken jack standard taken from scrap pile and reclaimed by electric welding process.

Statement showing comparative cost of reclaiming one jack standard as against purchasing new:

Cost new		$4.00	
Scrap value of broken standard		.30	
Net cost new			$3.70
Cost to Reclaim by Electric Welding Process:			
Labor:			
¾ hour, at 27½ cents per hour	$0.20		
Redrilling holes	.10		
		$0.30	
Material:			
1 pound of welding wire, at 5 cents per lb.	.05		
Electricity, ¾ hour, at 12 cents per hour	.09		
		.14	.44
Net saving by electric welding			$3.26

Figures 10 and 11 show feed-water pump reclaimed by electric welding process.

Statement showing comparative cost to reclaim by electric welding as against purchasing new:

Cost of new cylinders machined, ready to set up		$220.00	
Deduct for scrap value		9.00	
Net cost of new pump exclusive of working parts			$211.00
Cost to Repair by Electric Welding:			
Labor:			
Welding, 8 hours, at 27½ cents per hour	$2.20		
Boring out and applying bushing where broken	6.00		
		$8.20	
Material:			
7 pounds of welding wire, at 5 cents per lb.	.35		
Electricity, 8 hours, at 12 cents per hour	.96		
Brass bushing	1.75		
		3.06	
			11.26
Net saving by electric welding			199.74

Figure 12 shows wheel lathe cutting tool, 1¼-inch by 3-inch by 16 inch, made from tire steel with high-speed cutting point attached by electric welding process.

Statement showing comparative cost of tool made by above mentioned as against tool made entirely of high-speed steel:

Tool Made from High-Speed Steel:			
Labor:			
Forging and finishing complete		$0.60	
Material:			
18 pounds high-speed steel at 55 cents per pound		9.90	$10.50
Tool Made from Tire Steel Having High-Speed Steel Point Attached:			
Labor:			
Forging tool and point complete	.55		
Electric welding on point	.15		
Dressing off	.08	.78	
Material:			
18 pounds tire steel at .5 cent per pound	.09		
¾ pound high-speed steel at 55 cents per pound	.41		
¼ pound welding wire at 5 cents per pound	.01		
Electricity, ½ hour at 12 cents per kw. hour	.06	.57	1.35
Total initial saving			9.15

NOTE.—As this tool becomes shorter by grinding, new points will be required, and two points beside the first one applied will be necessary in order to get the same life as could be obtained from entire high-speed tool. These points are applied at a cost of .90 each for labor and material; thus the total final saving on tool amounts to $7.35

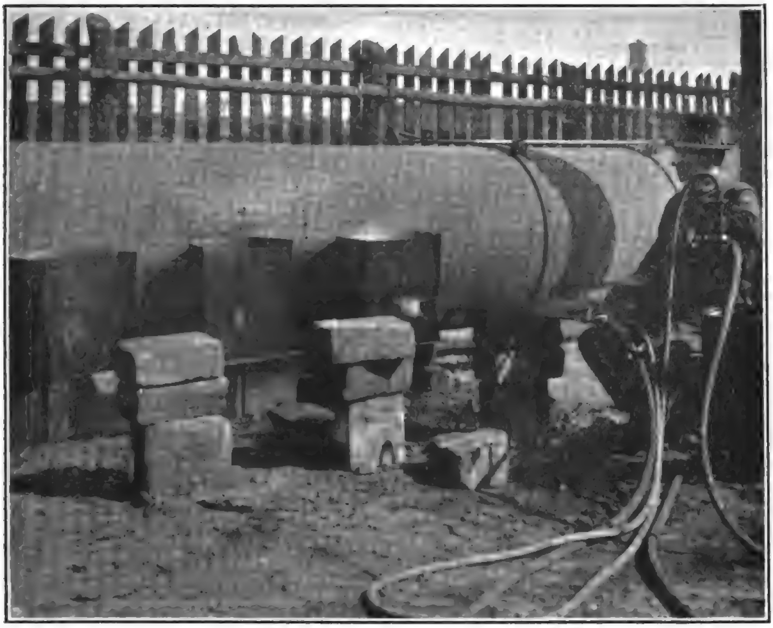


FIG. 1.

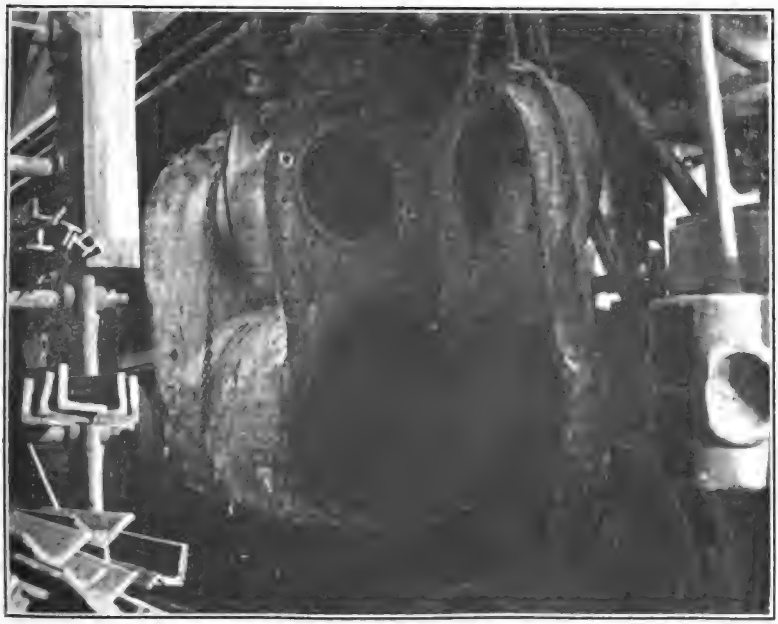


FIG. 2.

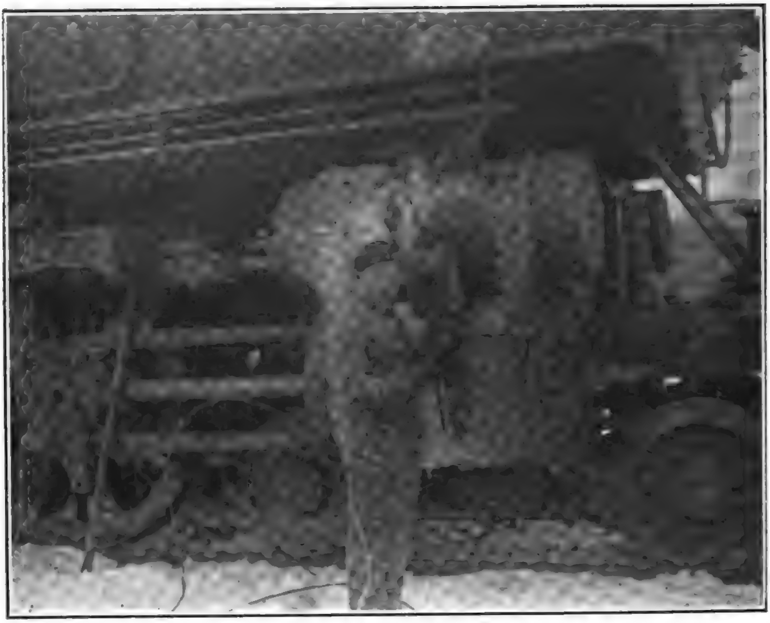


FIG. 3.

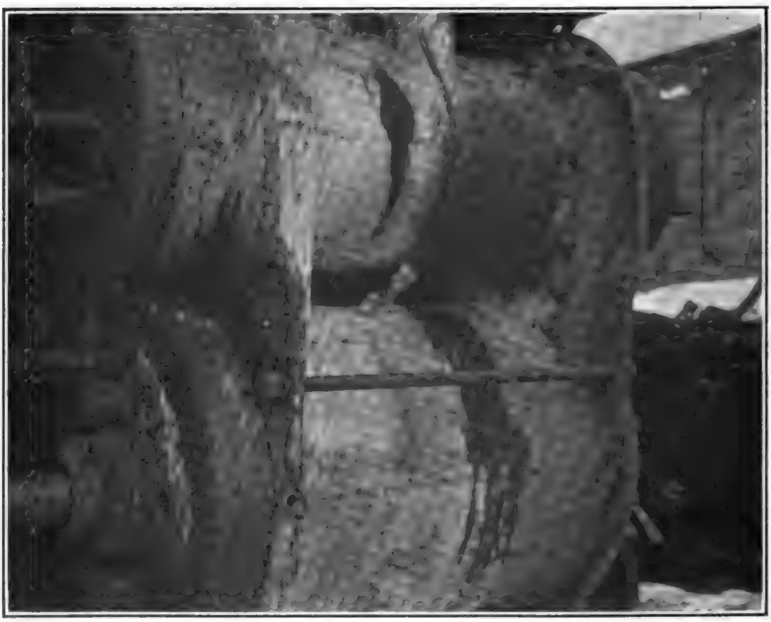


FIG. 4.

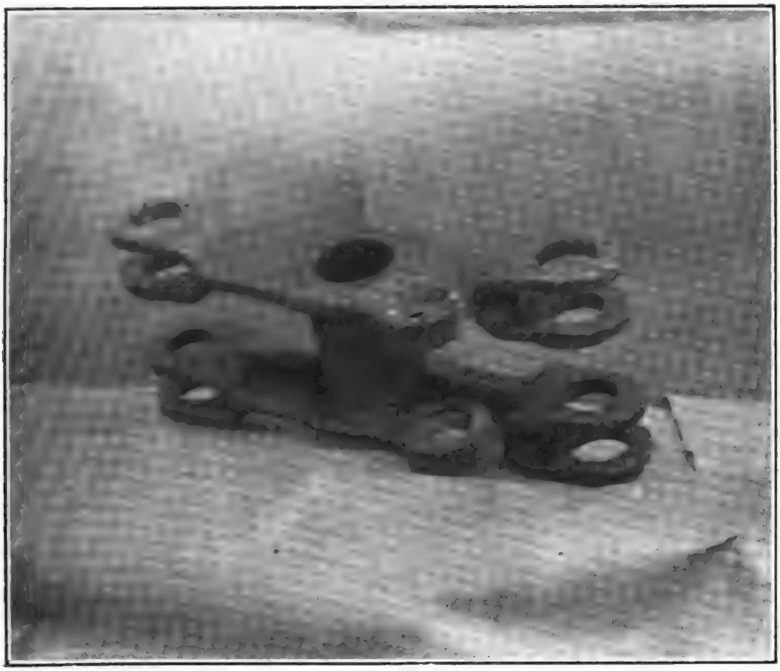


Fig. 5.

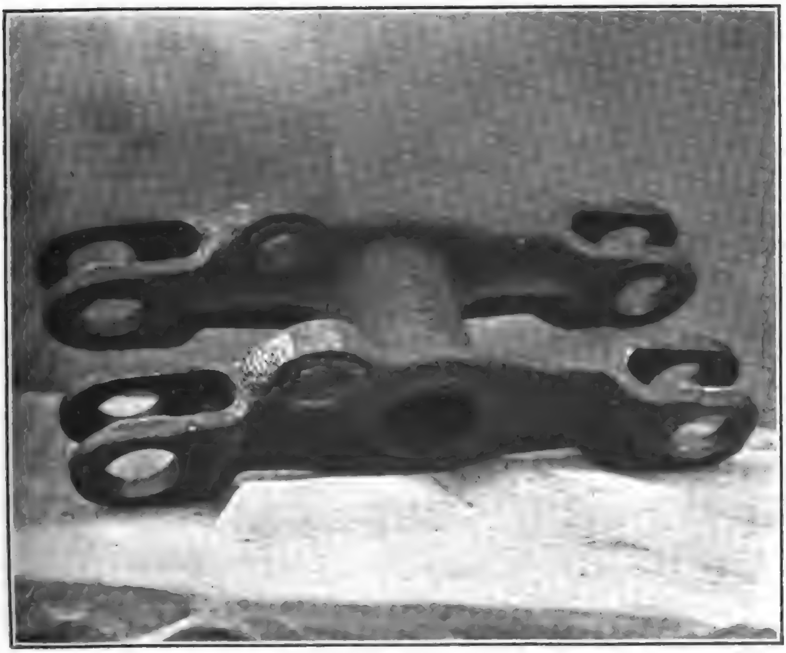


Fig. 6.

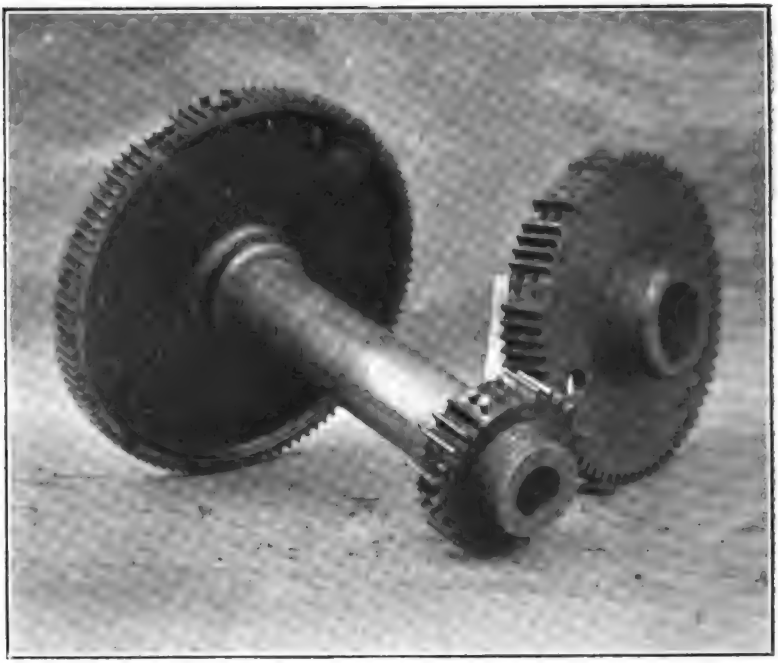


Fig. 7.

Fig. 8.

FIG. 9.

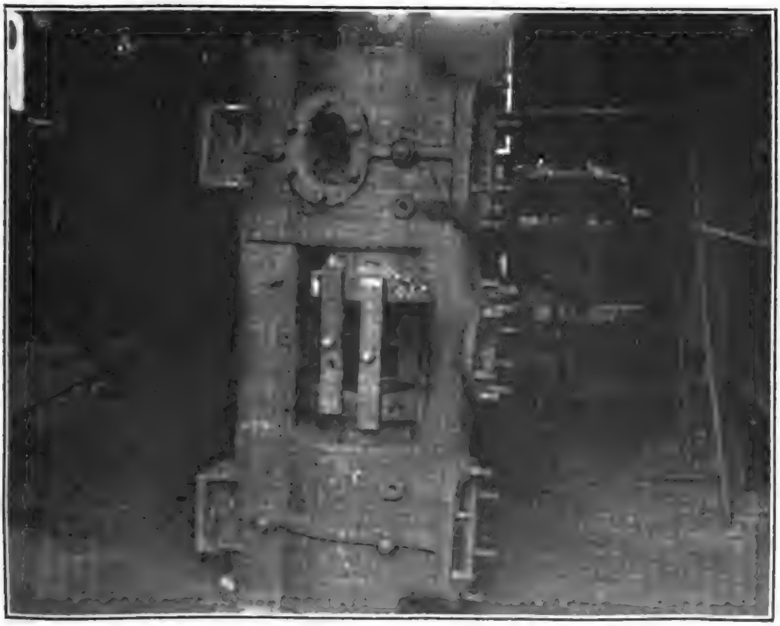


FIG. 10.

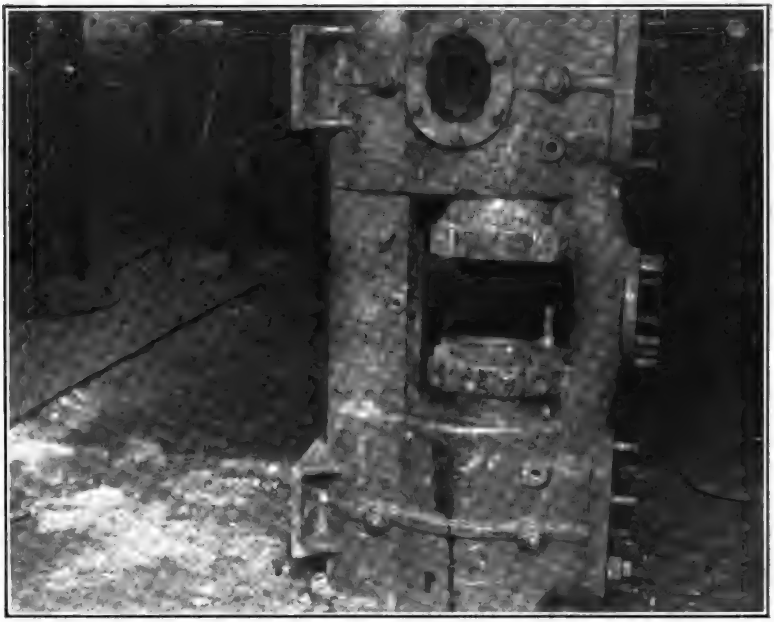


FIG. 11.

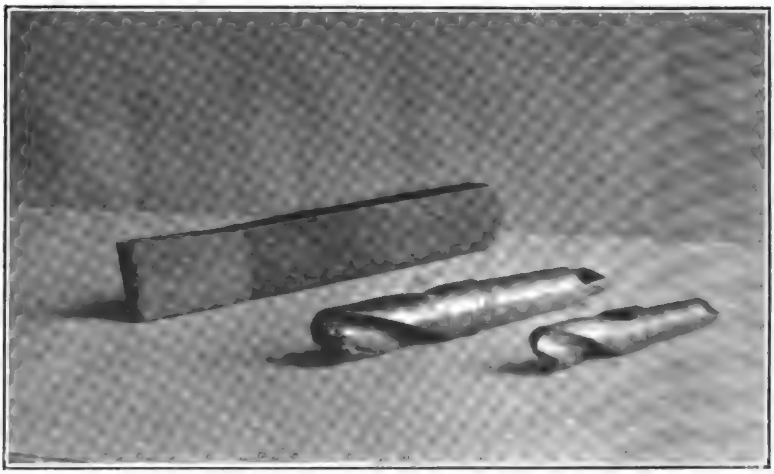


FIG. 12.

Short $2\frac{1}{16}$-inch carbon drill converted to high-speed drill by electric welding in an insert of high-speed steel at cutting edge.

It is estimated that the life of three inserts will be equal to that of a new carbon drill 12 inches in length.

Statement showing comparative costs:

One new $2\frac{1}{16}$-inch by 12-inch carbon drill			$4.50
Cost to apply High-Speed Insert to Old Drill:			
Labor:			
Preparing cutting edge for insert	$0.35		
Welding in insert	.10		
Dressing	.15	$0.60	
Material:			
2 ounces high-speed steel at 55 cents per pound	.07		
Welding wire	.01		
Electricity, $\frac{1}{3}$ hour at 12 cents per hour	.04	.12	
Total cost and labor and material		.72	
Three inserts			$2.16
Total saving by electric welding in inserts over cost of new drill			2.34

U. S. S. *WADSWORTH.*

DESCRIPTION AND TRIAL PERFORMANCE.

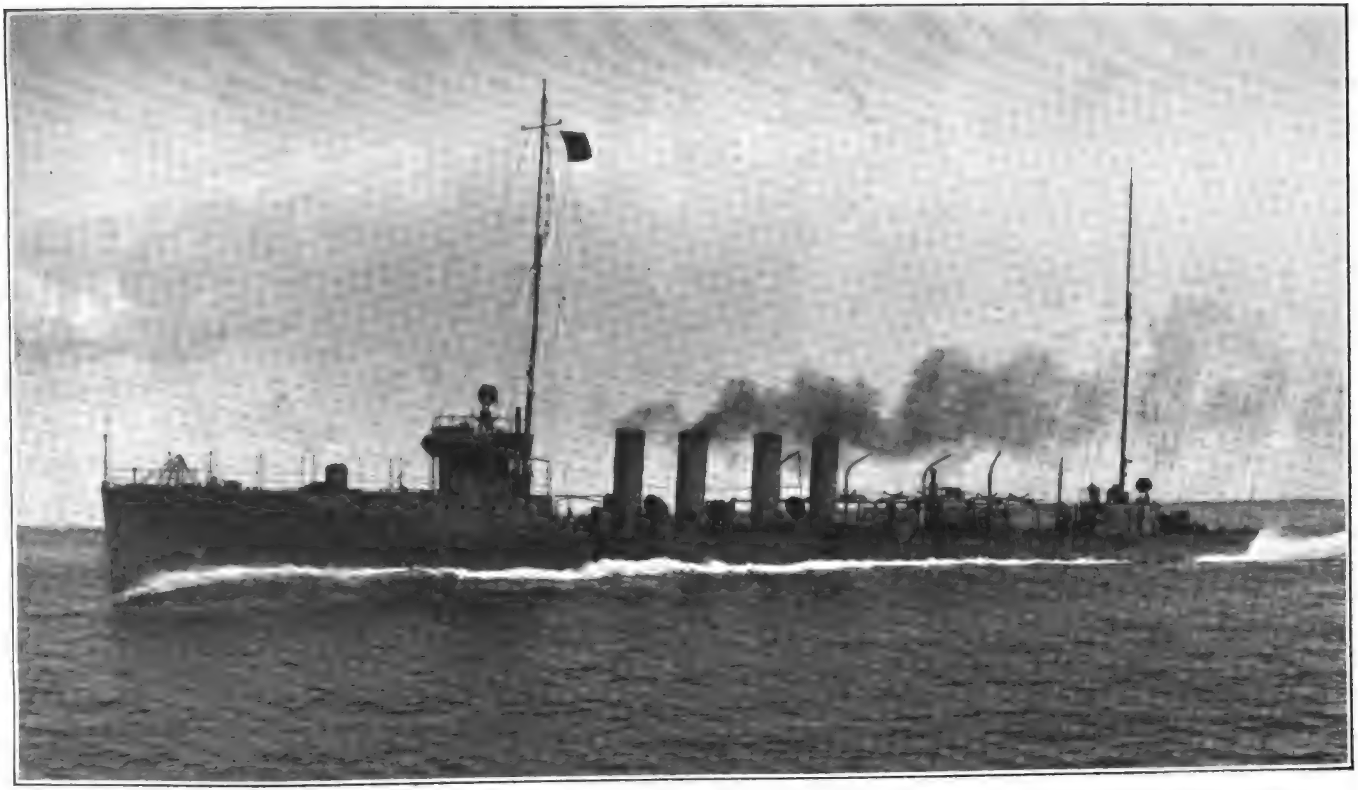
By Henderson B. Gregory, Associate.

The builders, Bath Iron Works, Limited, of Bath, Maine, have recently successfully completed the preliminary official trials of the U. S. S. *Wadsworth.*

Officially known as Torpedo Boat Destroyer No. 60, the *Wadsworth* is one of six boats authorized by an Act of Congress, approved March 4, 1913. The contract for her construction was signed October 15, 1913, the time of completion being twenty-four months and the price $884,000.00. She is a twin-screw vessel, driven by Parsons turbines, fitted with reduction gear, and designed for a speed of 30 knots, at a displacement of about 1,050 tons, with the turbines developing 17,500 shaft horsepower. Steam is supplied by four Normand water-tube boilers.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	310–0
on L.W.L., feet and inches	310–0
overall, feet and inches	315–3
Beam, extreme, feet and inches	30–6
on L.W.L., feet and inches	29–8¼
Ratio of length to beam (L.W.L.)	10.446
Draught to L.W.L., mean, feet and inches	9–4¼
Displacement corresponding, tons	1,050
per inch at L.W.L., tons	13.8
Area immersed midship section, square feet	210
L.W.L. plane, square feet	5,775
Wetted surface, square feet	10,000
Coefficient of fineness, block	0.44
midship section	0.75
L.W.L.	0.63



U. S. S. "WADSWORTH."

GENERAL DESCRIPTION OF HULL.

The vessel is of the customary raised-forecastle type. There are two masts, fitted with wireless, signal yards, etc., and four smoke pipes.

Forecastle Deck.—This deck, about 80 feet long, provides for the stowage of anchors, bow gun and pilot house. On top of the pilot house and extending to the vessel's sides is the bridge fitted with a steering wheel. Above the bridge is a searchlight platform, mounting a 30-inch searchlight.

Main Deck.—The main deck is continuous from stem to stern. Under the forecastle deck are the officers' quarters, pantry, galley, radio room, etc. Abaft the forecastle is a weather deck on which are located the battery, described elsewhere, and a deck house aft, containing the crew's toilet rooms, etc. On top of the deck house is a steering platform and a 30-inch searchlight.

Platform Deck.—This deck extends forward and aft of the machinery space only. Forward it contains the paint and oil room and crew's quarters, and aft are located crew's and petty officers' quarters, and steering gear.

Hold.—In the hold, forward and aft, are located trimming tanks, fuel-oil tanks, magazines and stores. The chain lockers are forward, and the machinery spaces occupy the entire mid-portion of the hold. A reserve feed tank is located in the forward engine room and fresh-water tanks in the forward fireroom, port and starboard. The engineer's store rooms are in the firerooms.

BATTERY.

There are four 4-inch rapid-fire guns in the battery, one on the forecastle, two on the main deck forward, port and starboard, at break of forecastle, and the other on the main-deck center line shaft of deck house.

The torpedo equipment consists of four 21-inch twin torpedo tubes mounted on the main deck, two port and two starboard.

There is a four-stage Ingersoll-Rand air compressor for

charging the torpedoes. It is located in forward engine room and has a capacity of 20 cubic feet of air at 2,500 pounds per square inch per minute. It is steam driven, the steam cylinder being 8 inches diameter and the compressor cylinders $9\frac{1}{4}$, $4\frac{3}{4}$, $3\frac{1}{8}$ and $1\frac{1}{4}$ inches diameter, respectively, for the four stages, with a common stroke of 5 inches. No accumulator is fitted.

BOATS CARRIED.

The following small boats are carried on skid beams above the main deck just abaft the midship frame. Davits are provided for launching the boats.

1 24-foot motor-sailing launch;
1 21-foot motor dory;
1 24-foot whaleboat;
1 14-foot wherry;
1 10-foot punt.

COMPLEMENT.

Commanding officer, . . .	1
Wardroom officers, . . .	4
Crew, ...	95
Total, ...	100

ANCHOR WINDLASS.

The windlass engine is located on the main deck just forward of the officers' quarters, with the capstan above on the forecastle. It is a double, vertical engine, made by the Hyde Steam Windlass Co., with cylinders 5 inches diameter by 4 inches stroke.

STEERING ENGINE AND GEAR.

The steering engine is of the Hyde Steam Windlass Company's double, vertical type, with steam cylinders $6\frac{1}{2}$ inches

diameter by 8 inches stroke. It is located on the platform deck in the steering-gear room aft.

In addition to the steam gear an emergency hand-steering gear is also provided in the steering-gear room.

FIRE MAIN.

The fire main extends throughout the machinery spaces on the port side close under the main deck beams. It is 3 inches outside diameter and is led forward and aft into the crew's quarters. The main is supplied by the three fire and bilge pumps and the oil-cooler circulating pump, as noted in Table I. Branches for the various fire plugs are taken off the main at convenient locations. All fire plugs are fitted for 1½-inch hose and are located as follows:

Plug No.	Location.
8, ...	Forecastle deck.
4 to 7 inclusive, . . .	Main deck.
1 and 3, ...	Platform deck.
2, ...	Forward fireroom.

In addition to the regular fire plugs above enumerated there is a 1½-inch hose valve on the discharge manifold of each fire and bilge pump.

Forward and aft there are 2½-inch connections from the fire main for magazine flooding and sprinkling.

A 1-inch connection is taken off the fire main aft for flushing out the stern-tube bearings, a full size branch being led to each bearing.

FLUSHING SYSTEM.

The flushing system is taken off the fire main direct. There is a 1½-inch connection forward supplying the officers' water closet and ship's galley, and one of 2½ inches aft for the crew's water closets and wash room.

FRESH–WATER SYSTEM.

Fresh water is carried in the ship's tanks, located in the forward fireroom, port and starboard. Each tank has a 2½-inch filling connection, fitted with hose valve at the ship's side. There is also a 2-inch filling connection from the distilling apparatus to each tank.

Located on the forecastle deck, immediately abaft the pilot house, is a small gravity tank for supplying the galley and officers' showers. The tank is fitted with steam coil to prevent freezing in cold weather. A hand pump is provided on main deck abaft the galley for filling this tank.

DRAINAGE SYSTEM.

A main drain is led throughout the machinery spaces and connected to the fire and bilge pumps by suctions the full size of the main. Macomb strainers are fitted in these suctions close to the pumps. The main is 4 inches outside diameter in the engine room and after fireroom, reducing to 2½ inches outside diameter in the forward fireroom. There are two full size bilge suction connections from the main drain in each fireroom and one in each engine room, fitted with plate strainers at the bilge ends.

For draining compartments outside of the machinery spaces there are 2½-inch outside diameter branches leading forward and aft and fitted with connections to the various compartments requiring drainage and the trimming tanks.

In addition to the drainage system proper, each engine room is provided with a 7-inch independent bilge-suction connection to the circulating pump for emergency use.

VENTILATION.

The officers' and crew's quarters are provided with a forced-ventilation system. There are two circuits, one forward and the other aft, each having a 2,500-cubic foot B. F.

Sturtevant fan, driven by a General Electric motor. All other spaces are ventilated by natural means, except the firerooms which, when closed, are ventilated by the forced-draft blowers.

HEATING SYSTEM.

The usual heating system, with brass-pipe coil radiators, is installed. Steam for the quarters forward is taken off the auxiliary steam line in the forward boiler room, and that for the after compartments from the auxiliary steam line in the engine room.

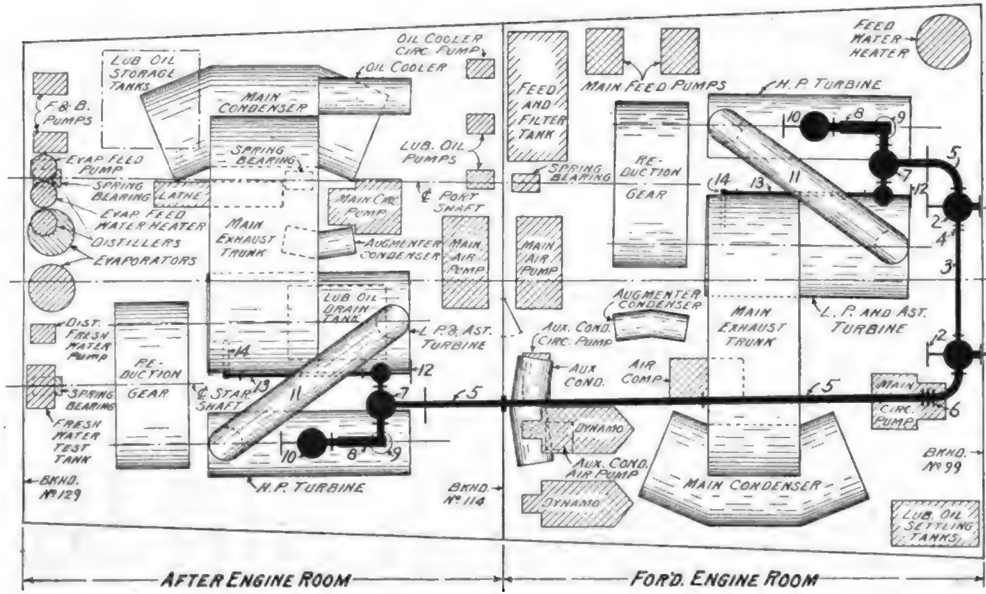
MAIN ENGINES.

General.—There are two engine rooms separated by a watertight bulkhead, as shown in Plate I. The propelling machinery consists of Parsons' geared turbines for a twin-screw drive. Each shaft is driven through gearing by one H.P. ahead and one L.P. ahead and an astern turbine; the latter is fitted in the same casing with the L.P. ahead turbine. The installation is designed to drive the screws at 450 r.p.m. when the main turbines are developing 17,500 S.H.P., corresponding to the contract speed of 30 knots.

Turbines.—The turbines are of the standard Parsons reaction high-speed type, the H.P. rotor operating at 2,494.5 r.p.m. and that for the L.P. at 1,509.8 r.p.m., when driving the screws at the designed speed of 450 r.p.m. The H.P. turbines are fitted with cruising stages and the L.P. turbines with reversing stages. The casings are of cast iron and the rotors forged steel.

The dummies and rotor-shaft glands are steam packed with the usual labyrinth packing, and the H.P. turbines are fitted with micrometer gear for measuring the dummy clearances. No micrometer gear is required for the L.P. turbines, which are fitted with dummy strips of the circumferential clearance type.

There is a main bearing at each end of each turbine for supporting the rotor, consisting of white metal-lined bottom brass and cap.



ITEM	DESCRIPTION
1	10¼" O.D. MAIN STEAM FROM BOILERS.
2	9½" MAIN STEAM STOP VALVE
3	7½" O.D. MAIN STEAM CROSS-CONNECTION
4	EXPANSION JOINT
5	10¼" O.D. MAIN STEAM TO H.P. TURBINE
6	EXPANSION JOINT
7	9½" H.P. TURBINE THROTTLE VALVE
8	MAIN STEAM STRAINER
9	9½" STEAM TO H.P. TURBINE CRUISING STAGES
10	9½" BY-PASS VALVE STEAM TO H.P. TURBINE 1ST STAGE
11	20½" H.P. TURBINE EXHAUST TO L.P. TURBINE
12	6" ASTERN THROTTLE VALVE
13	6½" O.D. ASTERN STEAM
14	ASTERN STEAM STRAINER

PLATE I - U.S.S. WADSWORTH - ARRANGEMENT OF ENGINE ROOMS.

All turbines are provided with thrust blocks at the forward end, of the usual Parsons type. The lower half is arranged to take forward thrust and the upper half thrust in the opposite direction.

Main Turbine Data.

Rotor drums:	Diameter.	Length.
H.P., inches	$20\frac{3}{4}$ and 21	$81\frac{3}{4}$
L.P., inches	42	$34\frac{5}{8}$
Astern, inches	26	$31\frac{3}{16}$

Number of expansions:	
H.P.	12 (4 cruising and 8 main.)
L.P.	8
Astern	5
Turbine casing, diameter, each expansion:	
H.P. cruising, inches	22, $22\frac{1}{2}$, $22\frac{5}{8}$ and 23
main, inches	23, $23\frac{3}{8}$, 24, $24\frac{5}{8}$, $25\frac{1}{4}$, $26\frac{1}{4}$, $27\frac{3}{4}$ and $29\frac{1}{4}$
L.P., inches	47, $48\frac{1}{4}$, $49\frac{3}{4}$, $51\frac{3}{4}$, 54, three of 59
Astern, inches	$27\frac{1}{2}$, 29, 32, two of 38
Length of casing for each expansion and diameter noted above:	
H.P. cruising, inches	$7\frac{7}{8}$, $7\frac{7}{8}$, $8\frac{7}{16}$ and $10\frac{11}{16}$
main, inches	5, $3\frac{3}{4}$, 5, $3\frac{15}{16}$, $5\frac{1}{4}$, $4\frac{7}{8}$, $4\frac{7}{8}$ and $5\frac{1}{16}$
L.P., inches	$2\frac{1}{2}$, $2\frac{1}{2}$, $2\frac{5}{8}$, $3\frac{3}{8}$, $3\frac{3}{8}$, $5\frac{13}{16}$, $6\frac{3}{16}$ and 5
Astern, inches	$4\frac{3}{4}$, $5\frac{3}{4}$, $5\frac{1}{4}$, $5\frac{13}{16}$ and $7\frac{11}{16}$
Rows of blading for each expansion:	
H.P. cruising, each	9
main	4, 3, 4, 3, 4, 3, 3 and 3
L.P.	2, 2, 2, 2, 2, 3, 3 and 2
Astern	4, 4, 3, 3 and 3
Length of blades for each expansion:	
H.P. cruising, inches	$\frac{5}{8}$, $\frac{3}{4}$, $\frac{15}{16}$ and $1\frac{1}{8}$
main, inches	1, $1\frac{3}{16}$, $1\frac{1}{2}$, $1\frac{13}{16}$, $2\frac{1}{4}$, $2\frac{3}{4}$, $3\frac{5}{8}$ and $4\frac{1}{8}$
L.P., inches	$2\frac{1}{2}$, $3\frac{1}{8}$, $3\frac{7}{8}$, $4\frac{7}{8}$, 6, $8\frac{1}{2}$, $8\frac{1}{2}$ and $8\frac{1}{2}$
Astern, inches	$\frac{3}{4}$, $1\frac{1}{2}$, 3, 6 and 6

Rotor shafts and bearings:	Length of bearing.	Diameter at bearing.	Diameter of axial hole.
H.P., inches	$5\frac{1}{2}$	$6\frac{1}{2}$	3
L.P., and astern, inches	$6\frac{1}{2}$	$6\frac{1}{2}$	3

Thrust bearings, each:	
Collars on shaft, number	13
Thickness, inch	$0\frac{5}{8}$
Distance between, inch	$0\frac{9}{16}$
Outside diameter, inches	$7\frac{1}{2}$
Inside diameter, inches	5
Number of shoes, top	12
bottom	12

TURNING GEAR.

A hand-turning gear is provided for each line of shafting. Each comprises a plug, fitted to squared end of axial hole of reduction-gear pinion shaft, the external end of plug being also squared for reception of ratchet wrench. Both ratchet wrench and plug are readily disconnected when not in use.

LIFTING GEAR.

Efficient lifting gear is installed for the main turbines. The lifting mechanism is hand operated.

REDUCTION GEAR.

The reduction gears are after the Parsons' design. Each unit consists of two forged-steel pinions, forged integral with their shafts, driving a main gear. The main gear is secured to its shaft by bolts, two collars being provided on the shaft for the purpose. It is of the built-up type, consisting of a forged-steel rim securely bolted to two steel-plate discs, forming a wheel, the whole being cross-braced by a cast-steel cone fitted between the discs and through bolted with the after disc to the rim and with the forward disc to its shaft collar. The gear and pinions have helical cut teeth. The pinion shafts are direct connected to the turbines, the H.P. turbine driving the outboard pinion and the L.P. turbine the inboard pinion. The gear shaft is coupled direct to the main shafting, and on its forward end is fitted the main thrust bearing, of the Kingsbury pivoted type.* The gear box consists of a rigid cast-steel bedplate, supporting the bearings, thrust, etc., and securely bolted to foundations built in the vessel. The entire apparatus is inclosed in a sheet-steel casing and provided with an elaborate system of forced lubrication for the shaft bearings and spray nozzels for the gear and pinions.

Plate II is reproduced from a photograph, taken aboard the vessel, of the port reduction gear with upper casing removed.

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* A complete description of this type thrust bearing, by Lt. W. W. Smith, U. S. N., Member, was published in vol. 24, page 1151, of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

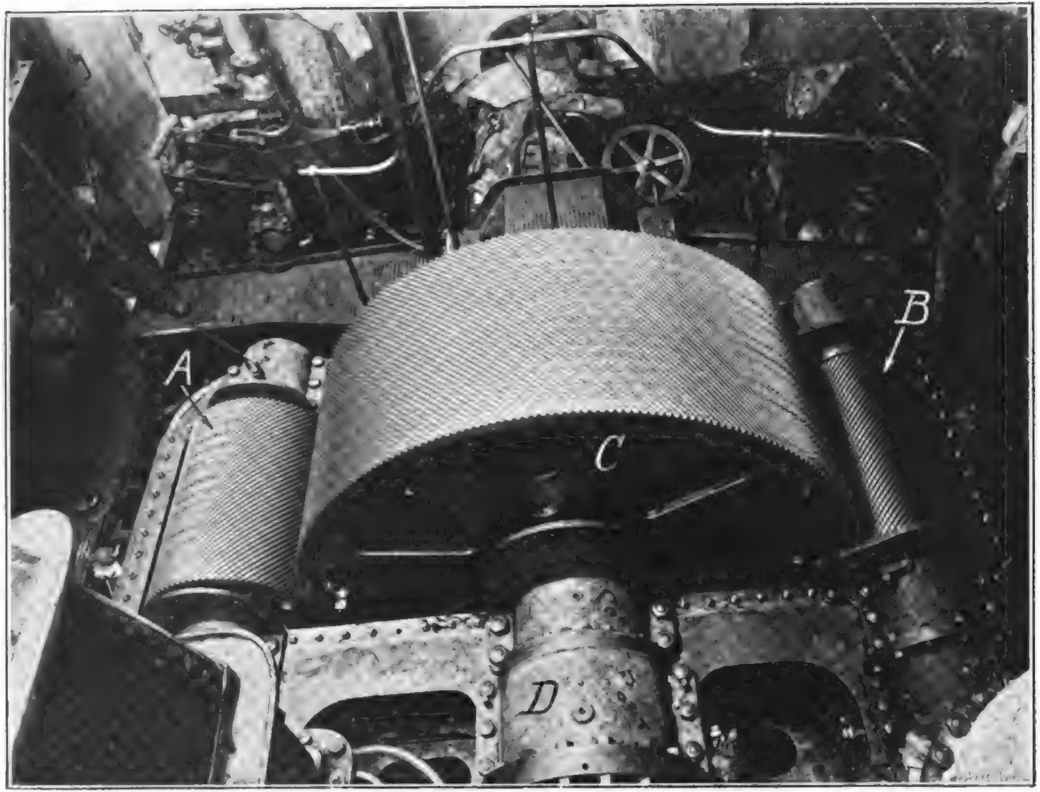


PLATE II.—PORT REDUCTION GEAR, LOOKING AFT. UPPER CASING REMOVED.

A. Inboard pinion, driven by L.P. turbine. *B*. Outboard pinion, driven by H.P. turbine. *C*. Main gear. *D*. Thrust box. *E*. Main shaft.

Reduction Gear Data.

	Wheel.	H.P. pinion.	L.P. pinion.
No. of teeth	255	46	76
R.p.m.	450	2,494.5	1,509.8
Diameter pitch circle, inches	66.227	11.947	19.738
over teeth, inches	66.467	12.187	19.978
Pitch, circular, inches	......	.8159	......
diametral	......	3.8504	......
Spiral angle, degrees and minutes	......	44–23.7	......

SHAFTING.

The starboard shafting is in three sections, consisting of one line shaft, one stern-tube shaft and one propeller shaft. The port side has four sections, an additional section of line shaft being necessary for extension to the forward engine room.

There are two stern-tube and one strut bearings for each line of shafting. The bearings are lined with lignum vitae. The shafts are composition bushed at the bearings, and within the stern tubes are covered with a seamless-drawn brass-tube casing, shrunk on.

Each section of line shafting is supported by two spring bearings, white-metal lined, of the self-oiling type.

The inboard coupling consists of a collar secured to the stern-tube shaft by four keys. Forward of this collar are two thin half collars fitted into a groove turned in the end of the shaft, to prevent the shaft from backing out, the whole being through-bolted to the coupling disc on the line shaft.

The outboard coupling is of the ordinary sleeve type secured to the shaft by four feathers and two cross-keys each.

Shaft Data.

Line shafts, diameter, outside, inches	10½
at journals, inches	10⅝
axial hole, inches	6¼
Stern-tube shaft, diameter, outside, inches	10½
axial hole, inches	6¼
Propeller shafts, diameter, outside, inches	10½
axial hole, inches	6¼

Coupling discs, diameter, inches	18
thickness, inches	$2\frac{1}{8}$
Inboard coupling, diameter of collar, outside, inches	18
inside, inches	$11\frac{1}{4}$
length of collar, inches	$7\frac{1}{4}$
thickness of half collars, inches	$1\frac{5}{8}$
Coupling bolts, number each coupling	8
diameter (taper)* at face of coupling, inches	$1\frac{15}{16}$
Outboard coupling, length of sleeve, inches	$36\frac{3}{4}$
diameter of sleeve, outside, inches	14
Spring bearings, diameter, inches	$10\frac{5}{8}$
length, inches	14
Forward stern-tube bearings, diameter, inches	$11\frac{1}{2}$
length, inches	$26\frac{1}{4}$
After stern-tube and strut bearings, diameter, inches	$11\frac{1}{2}$
length, inches	48

PROPELLERS.

The propellers are three-bladed. The port propeller is left-hand and the starboard propeller is right-hand. The blades and hubs are of manganese bronze and cast in one piece. The blades are true-screw machined to pitch.

Propeller Data.

	Starb'd.	Port.
Diameter, feet, and inches	7–7.75	7–7.5
Pitch, feet and inches	8–7.58	8–7.35
Ratio of pitch to diameter	1.128	1.29
Area, projected, square feet	25.416	
helicoidal, square feet	29.541	
disc, square feet	46.163	
Ratio, projected, to disc	0.55	
helicoidal, to disc	0.64	

MAIN CONDENSING APPARATUS.

Main Condensers.—There is one main condenser of circular section in each engine room. They are of the curved-tube type, with the tubes expanded in both tube sheets. The principal dimensions are as follows:

* Parallel bolts for inboard coupling.

Diameter of shell, maximum, inside, feet and inches	5-8½
Thickness of shell, inch	0½
tube sheet, inches	1⅛
Tubes, number	3,303
diameter, inch	0⅝
thickness, inch	0.049
mean length between tube sheets, feet and inches	10-0
Cooling surface, square feet	5,404.5
Main exhaust nozzle, area through, square feet	18.4
Diameter of air-pump suction, inches	11
circulating-water inlet and outlet, inches	20
auxiliary exhaust nozzle, inches	6
bleeder, forward condenser only, inches	2½

The main air and circulating pumps and their connections are given in Table I.

Vacuum Augmenter.—A Parsons vacuum augmenter is provided for each main condenser, consisting of a small condenser of the curved-tube type, steam jet and water seal, all connected to the main condenser and air-pump suction piping in the usual manner.

Each augmenter condenser is of the following principal dimensions:

Diameter of shell, maximum, inside, inches	21
Thickness of shell, inch	$0\frac{3}{16}$
Tubes, number	224
diameter, inch	0⅝
thickness, inch	0.049
Mean distance between tube sheets, inches	45.8
Cooling surface, square feet	139.6
Diameter of augmenter connection, inches	8
air-pump suction, inches	8
circulating-water inlet and outlet, inches	4

ENGINE-ROOM AUXILIARIES.

Pumps.—A complete list of the engine-room pumps and their connections is given in Table I.

Feed and Filter Tank.—A feed and filter tank of about 800 gallons gross capacity is located in the forward engine-room. The filter chamber is in the top of the tank, and so arranged that the entering water will flow over the top of a plate and

TABLE I – PUMPS AND CONNECTIONS – U.S.S. WADSWORTH.

No.	Pumps.	Size (Ins.)	Type	Suction Pipes From– Ins.		Discharge Pipes To– Ins.		Location
2	Main Air	(1) 14 x (2) 28 x 18	Twin, Vertical, Bucket, Single-Acting, "Warren"	11	Condensers, through Water Seal. Reserve Feed Tanks.	10	Feed Tank.	One in each Eng. Room
2	Aux Circ for Main Condensers	10" diam Impeller	Turbine Driven, Centrifugal, "Worthington."	14 7	Sea. Bilge	14	Main Condenser.	Do
2	Main Feed	15 x 10 x 16	Vertical, Piston, Double-Acting, Single, "Warren."	5½ 5½	Feed Suction Pipe and Reserve-Feed Tanks Water Seal Cross-Connecting Pipe.	4	Main Feed Discharge.	In Ford. Eng. Room.
2	Aux Feed	Do	Do	5½ 1½	Feed Suction Pipe and Reserve-Feed Tanks. Hose Connection.	4 1½	Aux Feed. Hose Connection.	One in each Boiler Room
2	Fire and Bilge	7 x 7 x 12	Do	4 4 4 1½	Sea. Drainage Bilge of Same Comp't. Hose Connection.	2½ 3½ 2½ 1½	Fire Main. Overboard. Distiller. Hose Connection.	In After Eng. Room
1	Do	Do	Do	3 3½ 1½	Sea Drainage Hose Connection.	2½ 3½ 1½	Fire Main. Overboard. Hose Connection.	In After Boiler Room
1	Aux. Circ for Aux Condenser	9 diam. Impeller	Electric Driven, Centrifugal	5	Sea	5	Aux Condenser	In Ford. Eng. Room.
1	Aux Air	6 x 10 x 8	Vertical, Piston, Double-Acting, Single, "Warren."	4½	Aux. Condenser.	4	Feed Tank.	Do
1	Evaporator Feed	4½ x 6 x 6	Do	2 1½	Sea. Distiller Circ. Disch.	1½	Evaporators.	In After Eng. Room
1	Distiller Fresh Water	Do	Do.	2½	Distiller Reservoir Tank.	2 2 2 1½	Reserve Feed Tanks. Ship's Tanks Cofferdam. Main Feed Tank.	Do
1	Oil Cooler Circ	7 x 7 x 12	Do	3	Sea	2½	Oil Cooler. Fire Main via Oil Cooler	Do
2	Lubricating Oil	6 x 7 x 8	Do	3½	Lub Oil Tank	3 1½	Bearings Settling Tanks via Forced Lub. System	Do
4	Fuel-Oil Service	5½ x 3½ x 8	Do	2½ 2½	Oil Storage Tanks Booster-Pump Disch	2	Burners	2 in each Boiler Comp't
2	Fuel-Oil Booster	4½ x 6 x 6	Do	3 3	Storage Tanks Deck-Hose Connection	2½ 2½ 2½	Service-Pump Supply Storage Tanks Deck-Hose Connection	One in each Boiler Room

leave through perforated plates in the bottom of the chamber after passing through the filtering material.

Auxiliary Condenser.—An auxiliary condenser, connected through the auxiliary exhaust pipe to all auxiliary machinery, is located in the forward engine-room. It is of the curved-tube type with tubes rolled in both tube sheets. The following principal dimensions are given:

Diameter of shell, maximum, inside, inches	22½
Thickness of shell, inch	$0\frac{3}{16}$
Tubes, number	334
diameter, inch	$0\frac{5}{8}$
thickness, inch	0.049
Mean distance between tube sheets, feet and inches	5–8
Cooling surface, square feet	310
Diameter auxiliary exhaust nozzle, inches	8
air-pump suction, inches	4½
circulating water inlet and outlet, inches	5

Feed-Water Heater.—There is a feed-water heater of the direct-flow type installed in the forward engine room, on the discharge side of both the main and auxiliary feed pumps. The heating agent is the exhaust steam, a back pressure being kept in the auxiliary exhaust line for this purpose by means of a spring-relief valve at each condenser connection, opening toward the condenser. The principal dimensions of the heater are as follows:

Diameter of shell, inside, inches	$23\frac{1}{2}$
Thickness of shell, inch	$0\frac{3}{16}$
Length between tube sheets, feet and inches	6–0
Tubes, number	509
diameter, inch	$0\frac{5}{8}$
thickness, inch	0.049
Heating surface, square feet	500
Diameter of feed inlet and outlet, inches	6
auxiliary exhaust nozzle, inches	8
drain, inches	2

Forced-Lubrication System.—The main bearings, thrust bearings, reduction gears and circulating-pump engine are provided with forced lubrication.

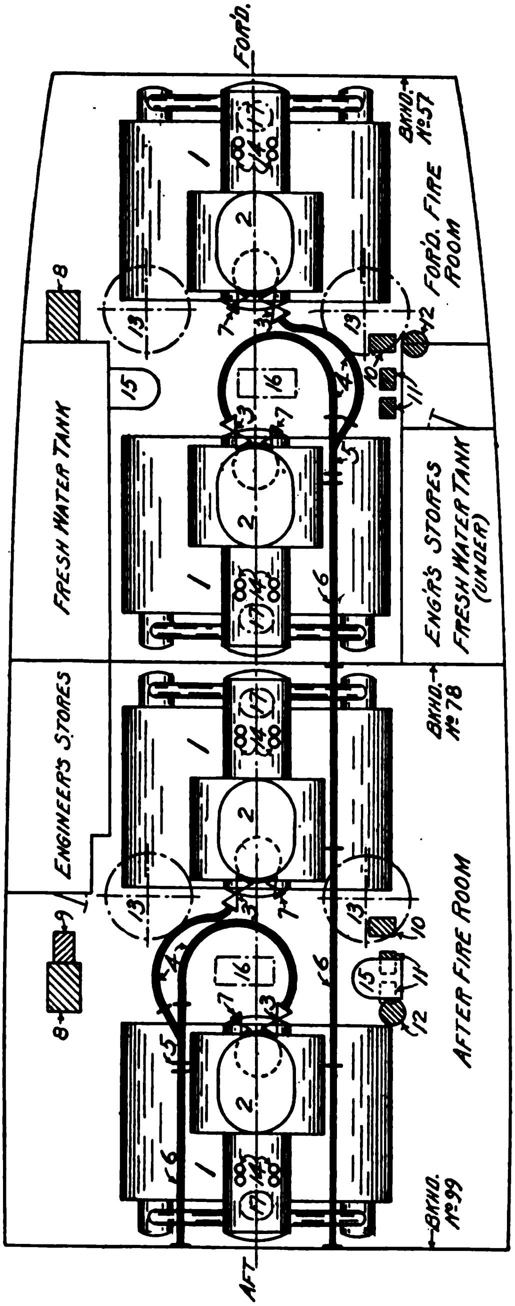
The system comprises two oil pumps,* one oil cooler of 291 square feet of cooling surface, one oil-cooler circulating pump,* one oil drain tank of 200 gallons capacity, an oil-settling tank of similar capacity and the necessary piping and fittings. Main storage tanks of 600 gallons capacity are also provided.

Lock cocks are fitted at each bearing for regulating the supply, and observation boxes for observing the oil flow from each bearing. A thermometer is also fitted, where practicable, to show the temperature of the oil leaving the bearings.

BOILERS.

There are four oil-burning Normand water-tube boilers, arranged in pairs in two separate compartments as shown in Plate III. They are designed to run the entire machinery

* See Table I.



ITEM	DESCRIPTION	ITEM	DESCRIPTION
1	BOILER	10	FUEL OIL BOOSTER PUMP
2	SMOKE PIPE	11	" " SERVICE "
3	7" MAIN STEAM STOP VALVE	12	" " HEATER
4	7½" O.D. MAIN STEAM PIPE	13	TURBINE DRIVEN BLOWER
5	MAIN STEAM EXPANSION JOINT	14	4" DUPLEX SAFETY VALVES
6	10¾" O.D. MAIN STEAM PIPE	15	AIR LOCK
7	3" AUX. STEAM STOP VALVE	16	FIRE ROOM HATCH
8	AUX. FEED PUMP	17	ESCAPE HATCH
9	FIRE AND BILGE PUMP		

PLATE III—U.S.S. WADSWORTH – ARRANGEMENT OF FIRE ROOMS.

plant at full power, with an average air pressure in the firerooms of about five inches of water.

Each boiler has an independent smoke pipe 20 feet high above the main deck.

Data for one Boiler.

Working pressure, pounds	260
Number of furnaces	1
Furnace volume, cubic feet	520
Area through smoke pipe, square feet	19.2
$\frac{\text{Smoke pipe area}}{\text{Furnace volume}}$	0.0368
Heating surface, square feet	5,375
$\frac{\text{Heating surface}}{\text{Furnace volume}}$	10.34
Oil burners, number	12
type	Modified Bureau of S. E.
External height, feet and inches	14–11$\frac{7}{16}$
width, feet and inches	16–02$\frac{1}{16}$
length, feet and inches	13–09$\frac{1}{2}$
Tubes, outside diameter, inches	1$\frac{1}{2}$–01$\frac{1}{4}$
thickness, inch	0.12–0.109
number	116–1,594
Downcomers, inside diameter, inches	10
thickness, inch	0$\frac{3}{8}$
number	2
Drum, upper, number	1
inside diameter, inches	44
thickness, shell, inch	0$\frac{11}{16}$
tube sheet, inches	1$\frac{3}{16}$
lower, number	2
inside diameter, inches	19
thickness, shell, inch	0$\frac{7}{16}$
tube sheet, inch	0$\frac{5}{8}$
Diameter of main steam stop valve, inches	7
auxiliary steam stop valve, inches	3
safety valves (two duplex Ashton), inches	4
main and auxiliary feed, stop and check valves, ins.	3
bottom-blow valves, inches	1$\frac{1}{2}$
surface-blow valves, inches	1$\frac{1}{4}$

FUEL–OIL SYSTEM.

The fuel-oil pumps and heaters are located in the firerooms as shown in Plate III.

The plant consists of two light-service booster pumps,*

* See Table I.

four heavy-pressure service pumps,* two oil heaters, and the oil-storage tanks in the forward and after holds, together with the necessary piping and fittings.

The booster pumps have suctions from all storage tanks and the deck connection for taking on oil, and discharge to all storage tanks, service pumps' suctions and the deck connections.

The service pumps draw oil from the storage tanks and the booster pumps' discharges, and deliver it to the oil burners on the boilers, via the oil heaters or bypasses. The burners, twelve per boiler, of the slightly modified Bureau of Steam Engineering type, are mechanical atomizers.

A small hand oil pump is installed in each fireroom for supplying oil to the burners when raising steam.

FIREROOM AUXILIARIES.

Forced-Draft Blowers.—Four forced-draft blowers are installed, two in each fireroom. The fans are of the Keith single-inlet type, mounted on vertical shafts in the base of the fireroom ventilators, from which the air supply is taken Each fan is driven by a 24-inch vertical Terry steam turbine located immediately below the fan.

The fan data is as follows:

Fan, diameter mean, inches	$41\frac{5}{8}$
width over all, inches	$13\frac{7}{16}$
number of blades	24
Diameter of inlet, inches	35
R.P.M. (designed)	1,600

Pumps.—Table I contains the list of fireroom pumps and connections.

MAIN STEAM PIPING.

The main steam piping is of seamless-drawn steel, with composition valves and fittings. The arrangement is clearly shown in Plates I and III, hence, further description is unnecessary.

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*See Table I.

AUXILIARY STEAM PIPING.

There is a 3-inch stop valve on each boiler for supplying the auxiliary steam line, which leads throughout the machinery spaces with connections to the various auxiliaries and a branch forward to the deck machinery, forward heating system and galleys. It is of same material as the main steam piping.

AUXILIARY EXHAUST PIPING.

An auxiliary exhaust pipe, of copper, extends throughout the machinery spaces and elsewhere as required for the various auxiliaries. Connections are provided for direct exhaust into either the main or the auxiliary condensers, the feed-water heater, or into the atmosphere through the after escape pipe, at will. There are also connections for admitting exhaust steam into the L.P. turbine's steam belt and into the fifth and eighth expansions of the H.P. turbine when desired.

MAIN AND AUXILIARY FEED PIPING.

The main and auxiliary feed pumps take their suctions from the feed suction main, which extends from the bottom of the main feed tank to the auxiliary feed pump in the forward fireroom. The main is 8 inches in diameter in the engine room, decreasing in size to 7 inches in the after fireroom and 5½ inches in the forward fireroom. The suction branches to all feed pumps are 5½ inches in diameter. All feed pumps are arranged to feed any boiler via the feed heater or bypasses. The main feed line is 6 inches in diameter from the feed-water heater to first branch connection to boilers, decreasing in size to 5 to 4 inches, as it leads forward in the firerooms. The discharge from each pump is 4 inches and the branches to all boilers are 3 inches each. Each auxiliary feed pump also has a direct connection to all boilers, through a 4-inch independent main between the two firerooms.

INTERIOR COMMUNICATION.

The customary engine-room and fireroom telegraphs, gongs, telephones, voice tubes, etc., are fitted for transmitting orders

and signaling to the machinery compartments and other parts of the vessel.

EVAPORATING AND DISTILLING APPARATUS.

The distilling apparatus is installed in the after engine room.

There are two evaporators and two distillers, with their accessories, arranged to operate in single or double effect. The plant has a combined nominal capacity of 5,400 gallons of water per twenty-four hours, with overload capacity 40 per cent. in excess of the nominal capacity when plant is clean.

Data for one Evaporator.

Type	Vertical, U-tubes.
Capacity per 24 hours, gallons	2,700
Diameter, inside, inches	34
Thickness of shell, inch	$0\frac{3}{16}$
Height overall, feet and inches	6–8
Tubes, number	50
diameter, inches	$1\frac{1}{8}$
thickness, inch	0.095
Heating surface, square feet	110
Diameter of steam nozzle, inches	2
vapor nozzle, inches	$3\frac{1}{2}$
feed valve, inch	1
blow valve, inches	2

Data for one Distiller.

Type	Vertical, straight tube.
Capacity per 24 hours, gallons	2,700
Diameter, outside, inches	$10\frac{1}{2}$
Thickness of shell (copper), inch	0.109
Length between tube sheets, inches	$30\frac{5}{8}$
Tubes, number	109
diameter, inch	$0\frac{5}{8}$
thickness, inch	0.049
Cooling surface, square feet	45.14
Diameter of circulating-water inlet and outlet, inches	2
vapor inlet, inches	2
drain, inches	$1\frac{1}{2}$

MACHINE TOOLS.

For performing minor repairs one small motor-driven engine lathe, of 12 inches swing and 6-foot bed, is installed in the

after engine room, together with all the necessary tools and attachments. The driving motor is a 1½-horsepower, direct-current, Reliance, adjustable-speed type, with speed ranges from 500 to 1,500 r.p.m.

ELECTRIC PLANT.

The dynamos are located in the forward engine room. The installation consists of two horizontal, compound-wound, direct-current, 25-kilowatt, General Electric generators, each driven by a Curtis steam turbine. Each generator will deliver at normal load 200 ampères of current at 125 volts, when running 3,600 revolutions per minute.

TORSION METERS.

Each line of shafting is fitted with a Gary-Cummings torsionmeter for ascertaining the shaft horsepower of the main turbines.

TRIALS.

The contract required the following trials for each vessel:

(*a*) A progressive trial over a measured-mile course at Rockland, Me., for standardizing the screws, extending from maximum speed (at least 30 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 30 knots.

Four fuel-oil and water-consumption trials, designated (*c*), (*d*), (*e*) and (*f*), of four hours' duration in the open sea in deep water, at an average uniform speed of 25, 20, 16 and 12 knots, respectively, as nearly as possible. The trials to be conducted as nearly as possible to service conditions.

(*g*) Trials to determine the ability of the vessel to back satisfactorily at full speed and at cruising speed, and one trial to determine the time required to bring the vessel dead in the water, and the distance head reached, when steaming ahead at 30 knots.

Fuel-Oil Consumption Guarantee.—The contractors guar-

anteed that the consumption of fuel oil per knot run, for all purposes, including that necessary for all auxiliaries in use on trial, shall not exceed 641 pounds at 30 knots, 403 pounds at 25 knots, 245 pounds at 20 knots, 233 pounds at 16 knots and 203 pounds at 12 knots, the consumption of fuel oil at these speeds to be determined by a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*), (*e*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

The trials were conducted June 21 to 24, 1915, inclusive. The contract speed was exceeded without effort, and the fuel-oil consumption guarantees were improved upon beyond expectation. Twenty-six runs were made on the standardization trial (*a*), as noted in Table II, which gives the data used in plotting the curves, Plate IV.

TABLE II—U.S.S. WADSWORTH
STANDARDIZATION TRIAL DATA—JUNE 21, 1915

No. of Run	Time on Course: Mins.	Time on Course: Secs.	Speed in Knots	R.P.M.: Star. Engine	R.P.M.: Port Engine	R.P.M.: Mean	S.H.P.: Star. Engine	S.H.P.: Port Engine	S.H.P.: Total
1	7	15.8	8.26	101.32	101.66	101.49	79	72	151
2	8	12.0	7.32	102.35	102.61	102.48	79	87	166
3	6	59.5	8.58	104.30	102.71	103.51	81	65	146
Mean of Group			7.87			102.49			157
4	3	23.1	11.07	151.08	151.05	151.07	277	245	522
5	4	44.3	12.66	156.34	154.82	155.58	331	240	571
6	5	23.5	11.13	150.15	149.98	150.07	244	233	477
Mean of Group			11.88			153.08			535
7	3	35.4	16.71	211.97	211.54	211.76	658	702	1360
8	3	49.8	15.67	210.29	210.02	210.16	727	682	1409
9	3	35.8	16.68	211.06	211.09	211.08	670	700	1370
Mean of Group			16.18			210.79			1387
10	2	57.8	20.25	272.54	272.62	272.58	1520	1578	3098
11	2	52.0	20.93	271.15	271.80	271.48	1493	1592	3085
12	2	56.3	20.42	272.97	272.99	272.98	1503	1638	3141
Mean of Group			20.63			272.13			3102
13	2	27.1	24.47	337.23	337.53	337.38	3237	3385	6572
14	2	29.3	24.11	337.39	337.27	337.33	3203	3285	6488
15	2	26.9	24.51	337.67	337.72	337.70	3170	3289	6459
Mean of Group			24.30			337.44			6502
16	2	13.7	26.93	388.09	387.57	387.83	5187	5343	10530
17	2	11.2	27.44	389.46	389.62	389.54	5195	5388	10583
18	2	13.2	27.03	388.00	388.36	388.18	5121	5425	10546
Mean of Group			27.21			389.02			10561
19	1	58.4	30.41	449.14	449.02	449.08	7608	7764	15372
20	2	00.0	30.00	447.52	448.22	447.87	7330	7812	15142
21	1	57.9	30.53	450.12	451.24	450.68	7657	7739	15396
Mean of Group			30.24			448.88			15263
22	1	55.5	31.17	485.22	485.35	485.29	8904	9115	18019
23	1	52.4	32.03	486.30	486.17	486.24	8408	9232	17640
24	1	55.6	31.14	484.90	484.43	484.67	9310	9059	18369
25	1	51.6	32.26	486.81	486.89	486.85	8668	9114	17782
26	1	55.7	31.11	487.29	487.33	487.31	9254	9186	18440
Mean of Group			31.64			486.02			18005

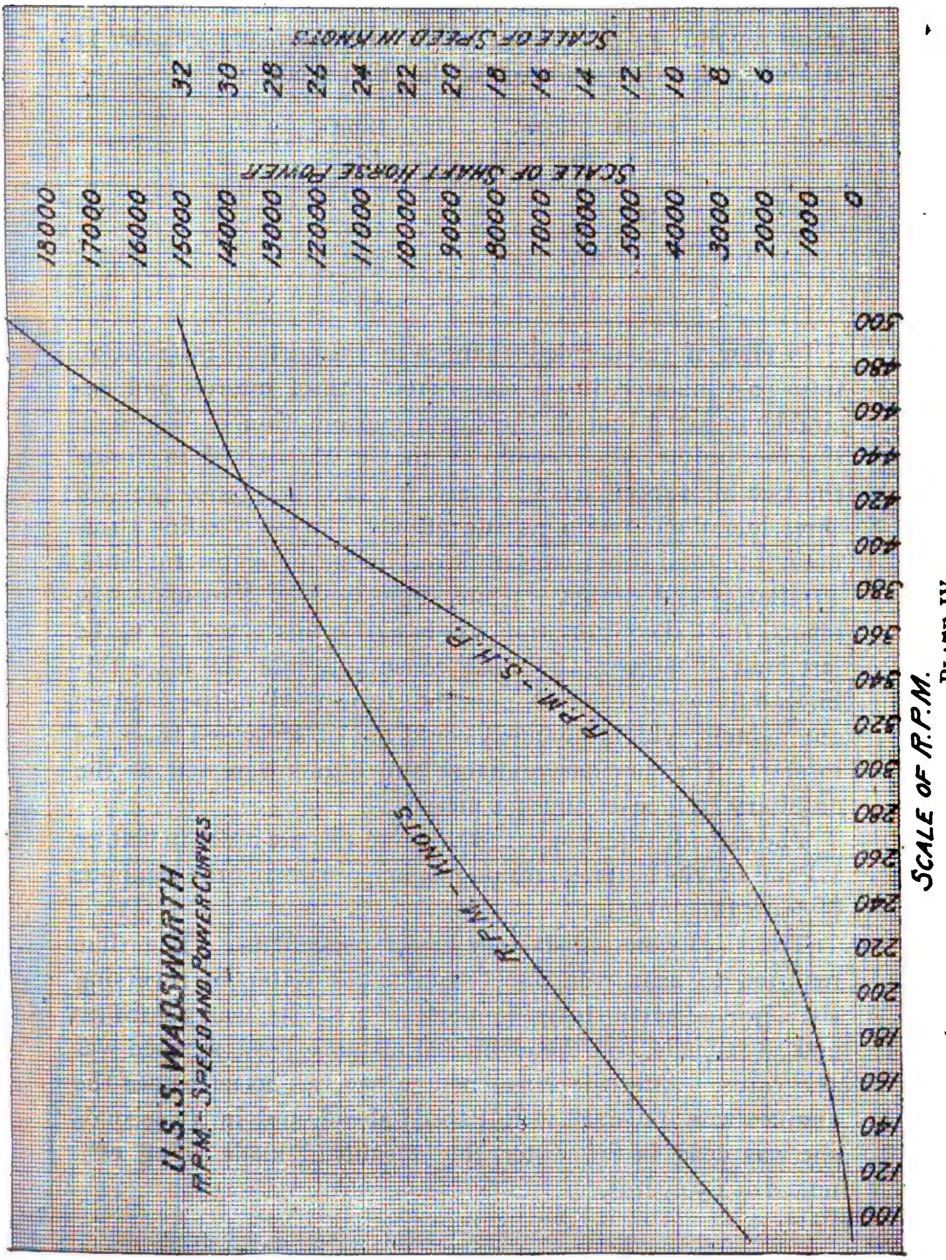

U.S.S. WADSWORTH
R.P.M. - SPEED AND POWER CURVES
SCALE OF SPEED IN KNOTS
32 30 28 26 24 22 20 18 16 14 12 10 8 6
SCALE OF SHAFT HORSE POWER
18000 17000 16000 15000 14000 13000 12000 11000 10000 9000 8000 7000 6000 5000 4000 3000 2000 1000 0
R.P.M. - S.H.P.
R.P.M. - KNOTS
500 480 460 440 420 400 380 360 340 320 300 280 260 240 220 200 180 160 140 120 100
SCALE OF R.P.M.

PLATE IV.

The official speed and revolution curve gave the following mean revolutions per minute of the propellers as necessary to attain the speeds stipulated for the other trials:

Speed, in knots.	R.p.m
30, ...	443.2
25, ...	349.8
20, ...	262.5
16, ...	208.5
12, ...	154.7

In view of the decided changes in the turbine installation from past practice, it was considered both of interest and desirable to get a line on the steam consumption of the main turbines only. To accomplish this it was agreed to divide the two high-speed trials, (*b*) and (*c*), into two parts each, of three hours' and one hour's duration. During the first three hours the total water consumption, including main turbines, auxiliaries and drains, was measured, and by agreement the official fuel-oil consumption was based on this period. On the last hour of the trial the water consumption of the main turbines only was measured, plus the steam used in the vacuum augmenters, which obviously could not be separated from the former. The auxiliary exhaust and drains were led to the auxiliary condenser and not measured.

The data for trials (*b*), (*c*), (*d*), (*e*) and (*f*) are given in Tables III and IV, the latter containing data taken during the last hour of trials (*b*) and (*c*).

Trials (*g*) were satisfactory in all particulars. The time required to bring the vessel dead in the water, by reversing the engines, when steaming ahead at 30 knots, was 3 minutes and 38 seconds, and the estimated distance head reached was 500 yards. After the vessel had gained full speed astern the engines were reversed to full speed ahead, and in 47 seconds she was dead in the water, the sternboard made being estimated at 150 yards.

TABLE III – TRIAL DATA – U.S.S. WADSWORTH

	4-HRS. FULL SPEED TRIAL (b)	4-HRS. 25 KT. TRIAL (c)	4-HRS. 20 KT. TRIAL (d)	4-HRS. 16 KT. TRIAL (e)	4 HRS. 12 KT. TRIAL (f)
FUEL OIL CONSUMPTION:					
B.T.U. PER POUND	19658.5	19658.5	19658.5	19658.5	19658.5
SP. GR. AT 60°F.	0.8726	0.8726	0.8726	0.8726	0.8726
POUNDS PER HOUR *	15364.09°	6845.41°	3115.4°	2072.16°	1397.21°
KNOT RUN	500.9°△	274.07°□	155.1°	129.48°	116.52°
AT	30 KTS. 473°	25 KTS. 275°	20 KTS. 153.5°	16 KTS. 129.48°	12 KTS. 116.52°
GUARANTEED AT	641°	403°	245°	233°	203°
BELOW GUARANTEE	168°	128°	91.5°	103.52°	86.48°
HOUR PER S.H.P.	0.955°§	0.919°+	1.063°	1.443°	2.536°
SQ. FT. OF H.S.	0.715°	0.318°	0.29°	0.386°	0.26°
SPECIAL DATA TAKEN IN FOR'D. ENGINE ROOM: (ALL READINGS INCHES OF MERCURY ABSOLUTE)					

LAND STORAGE OF BITUMINOUS COAL; THE EVER PRESENT FACTOR OF SPONTANEOUS COMBUSTION; AND A FEW FACTS AND SUGGESTIONS IN CONNECTION WITH SAME.

By Geo. R. Crapo, Paymaster, U. S. N., Naval Station, Key West, Fla.

Preface.—If the publishing of this article will result in the Departments of the Government making a careful investigation of every fire that occurs in coal-storage plants under their cognizance, and a record kept of same embracing the following: type of shed in which combustion took place, kind of coal, daily rise of temperature from time of taking in to time of combustion, length of time before combustion took place, whether coal was wet or dry when taken in, whether coal was green (freshly mined) or old, and the proportion of lump and dust in same, it is believed that valuable data will be collected, sufficient to save not only the Government but the commercial world large sums of money annually, and furthermore arrive at a condition whereby the Government will be able to store more coal in its present plants without fear of combustion.

The writing of this article is occasioned by the fact that from November 15th, 1914, to February 20th, 1915, a series of fires of a spontaneous nature, sixteen in all, took place in the coaling plant under my charge at the Naval Station, Key West, Fla.

The amount of worry, night calls and work, accompanied by a considerable expense in the continual handling of this hot coal for the purpose of cooling same, made it imperative that some study be given this matter that a re-occurrence might not occur. Not only from a standpoint of economy should this matter be made a subject of rigid investigation, but more vitally from a logistical standpoint, and for the following reason, which probably applies to other stations as well.

The plant at this station consists of two steel sheds: one 150 feet × 100 feet × 20 feet at the eaves, the other 250 feet

× 75 feet × 20 feet at the eaves. From the above dimensions, using 42.5 as the average density of coal, it is readily seen that the plant was built for the purpose of storing 15,883 tons of coal within ready reach of the conveyors for rapidity of handling. The best authorities now contend that coal cannot be safely stored at a depth of over 14 feet, and even at that depth a careful watch has to be kept on same.

This means that a plant designed to store coal in such manner that rapid discharge and loading of vessels can be effected, and with a capacity of nearly 16,000 tons, can only be utilized for the storage of approximately 11,000 tons.

If this condition applies to other naval storage plants (and I believe it does to a great extent) a large amount of government funds have been expended unnecessarily, as well as valuable property utilized for an unnecessary purpose, and, furthermore, an even more vital item presents itself in view of the fact that these plants being built for a certain capacity would probably be called upon in time of war to work to this capacity, and conditions might very probably arise that when this coal was very urgently needed, it would either be too hot to place in the vessels' bunkers with safety, or that the prolonged and continual process of heating had so exhausted its calorific qualities as to render it not good steaming coal. If not this, even the fact that the plant was not worked to capacity might seriously affect the situation by the lack of the amount of coal, the difference between what could be carried with reasonable safety and the plant's designed capacity.

Does it not appear that something is radically wrong; either our system of storage, or our lack of information on the subject, which if available might enable one to overcome existing conditions with the present plants?

One thing is certain that very little is known, or, if known, it has never been published; and, furthermore, it is believed that only a small percentage of coal combustions in land storage plants have ever been reported and made a subject of thorough investigation.

The existing published data on this subject is not complete, very inaccessible, and often contradictory. To quote Edgar Stansfield, M. Sc., McGill University, in his article on spontaneous combustion of coal, "A study of extensive literature bearing upon spontaneous combustion has shown that although much has been accomplished, a great deal more remains to be done. This literature is not easily accessible; and although there have been excellent summaries published, these chiefly refer to the question of shipment of coal cargoes, so that there does seem a need for a statement as to the known facts and present theories with especial reference to land storage."

In order that there will be no misunderstanding as to what takes place prior to ignition of coal, I wish to quote from the as yet unpublished notes of Mr. F. R. Wadleigh, Consulting Engineer, on this subject. "Spontaneous combustion of coal is a gradual cumulative absorption of oxygen, slow at first, but gradually gathering in amount with increasing temperature of the coal, until the ignition point is reached and the coal fires."

CONDITIONS UNDER WHICH COAL IS MORE LIABLE TO FIRE, AND SUPPOSED CONTRIBUTING CAUSES WHEREBY COAL IS EXCITED TO POINT OF IGNITION.

(*a*) If the surface of coal exposed is large.

(*b*) If air is supplied only at a rate sufficient to supply the required oxygen.

(*c*) If the heat generated cannot rapidly escape.

(*d*) When green coal (freshly mined) is stored with old coal.

(*e*) When coal both very fine (dust) and lump are stored together in a heap of any great height.

In addition to the above, Prof. Fischer, of Göttingen, has given to the world a very simple but practical method of determining in advance whether or not a coal in question can be safely stored, namely: "Coals which absorb bromine rapidly

are most liable to spontaneous ignition, and recommends the shaking of one grain of finely ground coal with 20 c.c. of a half normal solution of bromine for a period of five minutes. If the smell of the bromine has then disappeared, the coal is liable to oxidize rapidly and is not a safe one to store." Prof. Lewes also gives some further information as follows: "Coal that gains more than 2 per cent. in weight when heated to 250 degrees F. for 3 hours, is a very dangerous coal to store."

The above tests, being simple and requiring no elaborate or expensive apparatus, could be readily incorporated in the inspection and analysis of coals prior to loading, and forwarded in advance to the person to whom the coal is consigned, thus giving that person valuable information, whereby every precaution can be taken in coals that show by these tests an unusual tendency to develop heat.

SUPPOSED CAUSES.

(*a*) Escape of combustible gases when fresh surfaces are exposed.

(*b*) Absorption of oxygen from the air with consequent evolution of heat.

(*c*) Presence of moisture in abnormal percentages.

(*d*) External heat.

(*e*) Oxidation of sulphur compounds.

(*f*) Oxidation of bituminous shales.

(*g*) Climatic conditions.

(*h*) Depth to which coal is stored.

(*i*) Lack of proper ventilation.

The above ascribed causes are taken from the works of Perry Barker, Edgar Stansfield, McGill University, Messrs. Parr and Kressman, University of Illinois, and also from personal experience and observation.

Relative to the above-mentioned conditions under which coals fire most favorably, condition "A" is borne out both in theory and practice, as, regardless of technical expressions and phrases in which various authorities ascribe the reasons for

spontaneous combustion of coal, it resolves itself in plain English to "Absorption of oxygen," consequently the greater surface exposed the greater and more rapid the process of absorption.

Condition " B."—It being the air that supplies the oxygen, it stands to reason, and experience bears it out, that with air supplied only at a rate sufficient to furnish the required amount of oxygen for this process of absorption, the air currents, or ventilation, are insufficient to carry off the heat. In other words, I believe that insufficient ventilation is worse than none at all.

The remarks as to condition " B" are applicable to condition " C."

Condition " D."—Why this is I do not know, but time and time again in discharging a collier it has been necessary to place coal from this same collier both in empty bins as well as filling partially empty bins containing old coal, and invariably the mixture of old and new coal heats much more rapidly than the coal placed by itself, and, furthermore, I have seen the mixed bin fire where the solid bin remained almost as cool as when taken out of the vessel.

Condition " E."—Coal dust has now become accepted as not only an explosible agent but a carrier of the flames after ignition has taken place.

Reference is invited to page 13, Bureau of Mines Bulletin No. 20. " Mr. Henry Hall, English Mines Inspector,—experiments at St. Helens in an adit or drift—coal dust was laid on the floor and shots were fired at the face of the adit and the flames traveled its length of about 135 feet."

With a mixture of lump and dust the lumps create pockets, veins, run ways, etc., and each time a plant is placed in operation the continual trampling and shoveling of the stevedores excite the dust already deposited, as well as creating more dust, which arises temporarily and then settles back, filling all these interstices, and when coal has become heated to a point of ignition near one of these trains of dust, it is possible for

the flames to be carried rapidly to the end of the train, which might be some point quite distant, and the heat deposited in a favorable place for the starting of another fire. Furthermore, this fine dust clogs up all avenues of escape of generated heat and interferes with ventilation which might otherwise prevent this pile from heating to a danger point.

Relative to contributing causes, it has been stated elsewhere in this article that the existing data on this subject is often contradictory.

Causes "A" and "B."—These causes are agreed upon unanimously by the various authorities, and I have found nothing in experience and practice that offers proof to the contrary.

"C" Presence of Moisture in Abnormal Percentages.—It has been a generally accepted theory that excessive moisture in coal in land storage plants was a large factor in the process of heating. Quoting the Bureau of Supplies and Accounts in its letter 504–10a of March 4th, "Moisture, sulphur in the form of pyrites, etc., are considered to have a contributory effect toward spontaneous combustion."

Mr. Perry Barker, in his article on spontaneous combustion of coal, also ascribes this cause as an important factor; but, turning to the report of the Royal Commission appointed by the New South Wales Government, 1898, for the express purpose of settling this question of moisture, we find: "The experiment was made by placing 245½ tons of small coal in each of two similar bins, placed side by side under a roof, so that surface ventilation was secured. The sides of the bins were made of boards and no attempt was made to stop up the cracks between the boards, so that air could penetrate the coal with sufficient freedom.

"The coal in the two bins was of the same character, each wagon of coal being divided between the two bins. In one case the coal was loaded in an air-dry condition; in the other the coal was thoroughly wetted by playing a hose on it while it was being shoveled into the bin, and it was so saturated with moisture that the water ran out of the bottom of the bin. Each

bin was fitted with several thermometer tubes for ascertaining the temperature.

"The coal was bituminous and its analysis showed 14 per cent. ash, 0.5 per cent. sulphur. Daily observations of the temperatures were taken on a regular system.

"Results: The temperature of the dry coal rose steadily from 122 degrees F., its mean temperature after loading, till after about 60 days it reached 392 degrees F. in its central part, and was on the point of catching fire, when the experiment was stopped in order to avoid conflagration.

"In the wet coal the temperature of the central part only increased from 106 degrees F. to 138 degrees F. in 38 days, after which the temperature decreased.

"The inference is that wet coal is far less liable to spontaneously heating than dry coal."

I have handled coal both wet and dry; have discharged vessels in a downpour of rain, and while fires have at times broken out in both wet coal and dry coal, less trouble has been experienced with wet coal.

"D." As to whether or not this has a bearing on the question of spontaneous combustion I cannot say, as the plant at this station is situated where no external heat can be applied other than the heat of the sun, which I do not believe has any material effect, as most of the fires at this station have been in the coolest months; and, furthermore, while the fires referred to at the beginning of this article were constantly occurring, fire of a spontaneous nature also broke out at the plant at the Navy Yard, Boston.

"E." I have not noticed that this has any material effect, and am borne out in my opinion by Mr. F. R. Wadleigh, in his as yet unpublished notes on spontaneous combustion, as follows: "The long-held theory that sulphur or iron pyrites was the main or at least a largely contributing cause, has been pretty well proved to be incorrect and misleading."

"F." In connection with this cause I can only state that in the piles and near the seat of the fire, shale and the "motheri"

have been noticed in varying quantities when digging the fire out.

"G." Having only studied conditions as they exist at Key West, an almost tropical climate, I am unable to state, but the Government having coaling plants in all latitudes can by comparison probably arrive at this conclusion.

"H" and "I." It is believed that these two have a direct relation inasmuch as the height to which coal can be stored is dependent upon the completeness of the ventilation. In other words, I contend that coal can be stored in the open at a greater height than in the inclosed shed and be a safe coal to handle.

A concrete instance that seems to bear out this contention is that during the several winter months that the Michigan Northern Peninsular is closed to navigation large quantities of coal are stored for the purpose of operating the many mines in that section. This coal is stored in the open (with merely a roof to prevent the coal from being buried under a deep fall of snow) and piled much higher than would possibly be attempted in a shed, and yet it is very rare indeed where one reads of a spontaneous combustion in these piles.

RECOMMENDATIONS AND SUGGESTIONS BASED UPON THE FOREGOING FACTS AND CONDITIONS.

It is believed that an open shed with just enough superstructure to support the overhead mechanical devices for rapidity of handling of coal is far superior and safer than the closed type. It is possible that a light roof that can be easily opened and shut might be an advantage. It is especially so at this station where we are so dependent upon rain for our water supply, as these roofs are great collection agencies for water.

In this connection I wish to state that facts (local) do not bear out the statement of Commodore W. H. Beehler appearing some time ago in the "Naval Institute," "It is claimed that if a ton of fine bituminous coal be spread out on a concrete pavement and exposed in open air in this climate for one

year, it will lose all its calorific properties. The gases are simply free to evaporate from the coal, and when the coal has lost all its gases it will have lost all its heat units and be simply coke."

There is a marked difference physically between coal and coke, and it is doubted very much if the simple exposure to the elements will make this transformation without the aid of extreme heat.

The Florida East Coast Railway, The Peninsular and Occidental Steamship Company, and the Mallory Steamship Company, as well as several commercial concerns, keep varying quantities of coal at Key West in the open, much of it that I have seen having been stored here considerably over one year without any change in appearance, and this coal is still being used for steaming purposes with good results. Probably some of the calorific properties have been lost, but not any great amount, unless this coal has been subjected, as before said, to extreme heat; in fact, to approximately the ignition point.

Regarding the mechanical appliances for handling coal, many plants use the overhead-conveyor system, the apron being from 45 to 60 feet above the floor of the shed. In many, many instances I have seen buckets containing from two to five tons dumped from that height to a concrete (or other) hard surface used as flooring.

This is a dangerous procedure for two reasons: 1st, The coal in its fall generates some additional heat, and before this heat can escape another bucketful is dropped upon the first, and so on, so that when your pile reaches 15 or 20 feet in height the original heat of the coal is augmented quite considerably. 2d, The fall from this height breaks a large percentage of the lump and creates more fine coal or dust, which in itself is a dangerous element.

In every instance reduce the amount of fall by lowering your bucket before tripping.

If one is obliged to use a closed shed, keep it as open as pos-

sible, and when storing coal have as many air passages as possible between and around the piles.

If possible when working the plant, devise some sprinkling device to prevent the rising and re-settling of fine dust.

Install tubes in all piles, through which thermometers can be lowered, and take daily readings of same, and if time will permit, take readings of temperature of coal in the various hatches of the collier prior to discharge. This reading makes a very fair basis on which to compare the subsequent heating.

Too much dependence, however, should not be placed in the temperature readings, as it very frequently happens that that portion of coal that fires first may not necessarily be the portion which indicated the highest thermometer reading, as it is possible, and often does happen, that the portion first heated may get a draft of air in consequence of the heating and the heat moved and deposited elsewhere. The accompanying photograph will illustrate an instance in the recent fires at this station. The pile, part of which is shown in the photograph, developed signs of heating to such an extent that the top of the pile was removed and the danger was thought overcome. This was late in the afternoon; the next morning, at some distance from the above referred to hot spot, and where the coal was only 8 inches in depth, as indicated in the photograph, fire was burning brightly like an open grate. I can only attribute this to one of two reasons: The hot portion getting a draft of air in consequence of its heating and removing the heat to this spot, or, a mixture of old and green coal with widely different analyses.

The large majority of fires at this station have come to a point of ignition where the coal is either in contact with a wall or around a stanchion and almost invariably where the coal was not deeply piled.

The promiscuous use of water in extinguishing fires in a coal pile is practically of no value and often misleading as to the results; furthermore, a coal pile will often heat almost to the point of coking and then subside, but if water is used at

this state the coking will take place at once and coal will be lost that otherwise could be saved.

The digging out of the heated portion is the only safe and sure way, and, if after segregation from the rest of the coal it is found that the coal is on fire, hollow out the top of the pile (like the crater of a volcano) and, after putting the water on the sides, fill the crater with water; the water seals up the pile and the gases being unable to escape bursts the pile open and the application of water can then be made and the fire extinguished in short order.

In addition to the thermometers above referred to, iron or steel rods run through the piles at various angles often reveal the extent of the most heated portion, and their constant use (probing) is recommended in addition to the thermometers.

Separation of lump from fine coal by screening is recommended whenever possible, using the fine coal first and reserving the lump.

The net loss by these sixteen fires was less than 10 tons of coal, and it is believed that strict adherence to the above *established* theories and practice was the reason for this very inconsiderable loss.

Inasmuch as the Government has large amounts of coal in storage under varying climatic and storage conditions, an excellent opportunity is presented whereby the world at large may be benefited, if every fire is reported and a personal rather than theoretical investigation made, going into all possible conditions and causes. These results, if properly recorded and filed, may in time lead to the solution of this problem, and the Navy Department be enabled to work their many plants to capacity without fear of disaster, and both the several branches of the Government and the commercial world relieved of this continual annual loss, together with the worry and work contingent upon same.

In conclusion, I wish to state that I consider this subject of sufficient importance to continue into its study, and for this

reason I would recommend to any reader interested in the foregoing remarks to read the following:

"The Iron and Coal Trades Review" of March 13, 1914.

"Deterioration and Spontaneous Combustion of Coal," by Perry Barker-Arthur D. Little, Inc., Chemists and Engineers, 93 Broad Street, Boston.

"Lloyds' Register of British and Foreign Shipping." Report by their chief engineer surveyor on spontaneous combustion of coal.

"Spontaneous Combustion of Coal," by Edgar Stansfield, M. Sc., McGill University, Montreal.

University of Illinois "Bulletin No. 46."

"The Spontaneous Combustion of Coal," by Messrs. S. W. Parr and F. W. Kressman.

Report to Messrs. Castner, Curran and Bullitt on fires at coal sheds, Key West, Fla., 1900–1901.

Bureau of Mines Bulletin No. 20, on "Coal Dust as an Explosive Agent."

I also wish to thank Mr. F. R. Wadleigh for many valuable suggestions, and also his aid in helping me to secure numerous articles on this subject that otherwise would have been unattainable.

NOTES.

IRON A FACTOR IN THE WORLD'S PROGRESS.*

BY JOHN BIRKINBINE, CONSULTING ENGINEER, MEMBER OF THE INSTITUTE.

The European war, while drawing heavily upon the resources affecting industries of contending nations, also seriously influences conditions in countries throughout the world not engaged in hostilities, threatening lasting effect and possible future changes in many phases of development. For this reason a discussion is offered of the relative positions of the leading countries which produce pig iron or supply the requisite raw materials for its manufacture, and to indicate past as well as present conditions a period of at least four decades has been chosen as covering practically the era of greatest industrial growth.

Prognostications as to future conditions are not attempted, but analyses of the data presented may suggest possibilities for nations now prominent or others wherein the development or utilization of the above commodities is less pronounced, while similar studies of other industrial specialties may indicate even greater changes.

Iron ore and coal hold leading rank among mineral resources whose development has been and is coincident with the world's progress, for countries possessing these abundantly, or utilizing them liberally, are the dominant nations.

The United States produces more coal, more iron ore, and more pig iron than any other nation, and her strongest competitors are the European countries now at war, for they are each an important factor in the supply of some or all of the above, and, together with the United States, are depended upon for

85 per cent. of the iron ore,
96 per cent. of the pig-iron supply of the world,
89 per cent. of the coal.

They are also the most active producers of varied forms of iron and steel manufactures, a large proportion of their inhabitants being skilled and efficient in mine, furnace and mill work.

A disturbance as momentous as that produced by the European conflict is far-reaching in its effect upon industrial progress, and its influence may be expected to continue long after peace is declared; for animosities do not cease with the suspension of active hostilities.

Consideration of iron as a factor in the world's progress necessarily is dependent on statistics, but, to relieve the discussion of voluminous tables, charts are presented on which are traced the conditions prevailing since 1870 in iron ore, coal and pig-iron productions of important nations. The quantities recorded are in long tons (2,240 pounds), except for countries using the metric system, where the amounts are given in metric tons (2,204 pounds).

Iron Ore.—Plate A illustrates the annual production of iron ores in the countries which have been or are prominent in supplying this mineral. The ordinates indicate in tons (the space between two longitudinal lines

* Communicated by the Author.

representing five million tons) the outputs of each country, designated by a particular form of line; the abscissæ show the years. By tracing the "curve" for any country, its production in a given year is found, or its contemporaneous position with other iron-ore producing nations is demonstrated. Thus Plate A shows that Great Britain was the largest contributor of iron ore until 1889, when the United States took first place, and since 1895 Germany has exceeded the output of Great Britain. The contemporaneous contributions of the three nations were practically equal in the years 1893 and 1894, subsequent to which time the advance of Germany and of the United States has been more pronounced than that of

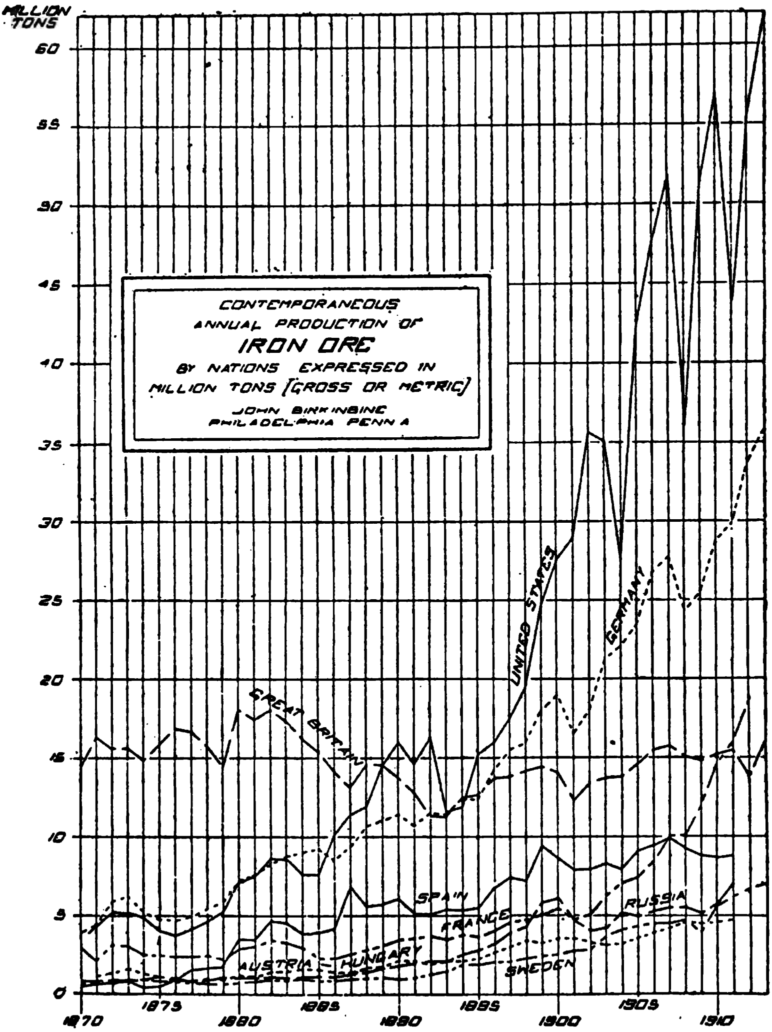


PLATE A.

Great Britain. In late years the iron-ore output of France has materially advanced. It will be noted that the maximum annual productions of iron ore, were for the United States 1913, Great Britain 1882, Germany and Luxemburg 1913,* France 1912, Spain 1907, Russia 1911 (?), Sweden 1913, Austria-Hungary 1911 (?).†

In addition to the countries designated in the graphic representation, the islands of Cuba and Newfoundland are each supplying about one and a half million tons of iron ore annually, and smaller amounts are mined in Canada, China, Greece, Italy, Algeria, Belgium, Tunis, Australia, India, Japan, Norway, Mexico, Brazil, Chile and Korea. Iron ore exists in minable quantities in most of the geographical divisions of the earth, and in some of these there are known reserves of enormous extent and of excellent quality.

The majority of iron ores obtained in Spain, Sweden, Russia and the United States average higher percentages of iron than those in Great Britain, Austria-Hungary, Germany and Luxemburg, France and Belgium. Hence the relative weights of iron ore obtained in the different countries may not be considered as equitably comparative, for two tons of ore containing 60 per cent. iron will produce as much metal as three tons of 40 per cent. ore. The world's output of iron ore approximates 157,000,000 tons, the United States furnishing 36 per cent., and the nations now actively engaged in hostilities in Europe supplying an aggregate of 49 per cent. of the total iron-ore product of the world. This resource is a most important factor, and it is asserted by some competent to judge that the control of the iron-ore deposits of what was then eastern France and Luxemburg had a potential influence in the Franco-Prussian war of 1870, and is a factor in the present conflict.

Pig Iron.—The graphic statement, Plate B, illustrates by ordinates, each space between horizontal lines representing 1,000,000 tons, the status of various countries as pig-iron producers subsequent to 1869, and indicates by "curves" the annual contribution of each nation in tons.

The diagram shows that England, which had long been preëminent in pig-iron manufacture, was outstripped by the United States in 1890, and that in 1903 Germany's product also exceeded that of Great Britain. Plate B, read in connection with Plate A, demonstrates that some important pig-iron producing countries depend to a greater or less extent upon other nations for iron ore, a feature which is later discussed.

Sweden, long celebrated for the excellence of its pig iron produced with charcoal as fuel, is included in Plate B, although its output is below, while those of all other nations shown on the diagram at present exceed 1,000,000 tons of pig iron annually. It will be noted that the maximum production of pig iron in each of the nations discussed was obtained within the past four years.

Among the countries supplying less than 1,000,000 tons of pig iron per year are: Canada (contributing over a half million tons), Spain, Italy, Japan, China, India and Mexico.

The production of pig iron indicates that various commercial forms of iron and steel into which it enters are also locally fabricated. The employment of liquid metal from blast furnaces for steel conversion, and of continuous processes of manipulation used in the manufacture of merchantable metal, causes the district which produces pig iron to attain prominence in supplying various forms of iron and steel, thereby advancing it as an important industrial factor.

To obtain pig iron, ores, fuels, fluxes and air are required; the last named is sufficiently abundant to cause no anxiety, and fluxes represent a minor part of the "charge." But the average ton of pig iron demands

* The statistics of Luxemburg, which supplies liberal quantities of iron ore and coal, have been officially combined with those of Germany.

† (?) intimates that no authentic records are available subsequent to the date mentioned.

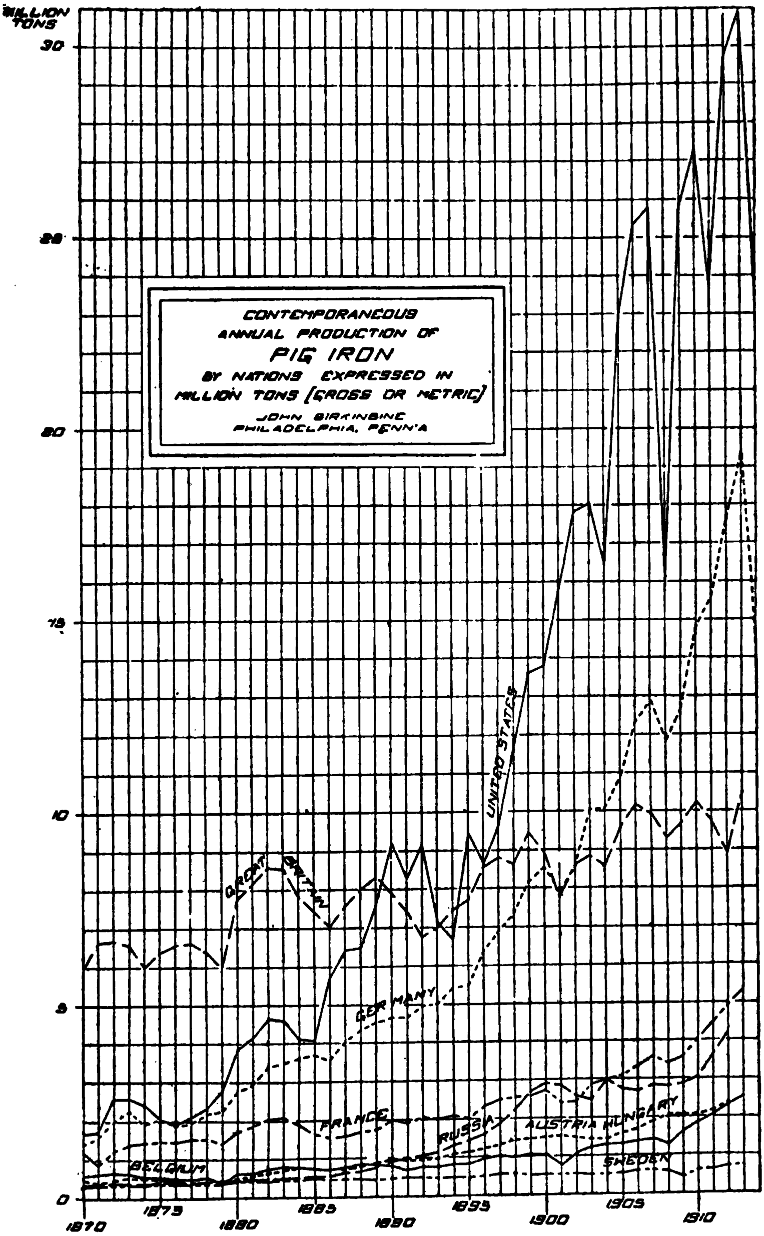

MILLION TONS
30
25
20
15
10
5
0
CONTEMPORANEOUS ANNUAL PRODUCTION OF PIG IRON BY NATIONS EXPRESSED IN MILLION TONS [GROSS OR METRIC]
JOHN BIRKINBINE PHILADELPHIA, PENN'A
UNITED STATES
GREAT BRITAIN
GERMANY
FRANCE
RUSSIA
AUSTRIA HUNGARY
BELGIUM
SWEDEN
1870
1875
1880
1885
1890
1895
1900
1905
1910

PLATE B.

for its production the consumption of two tons of ore and a ton or more of satisfactory fuel. Hence the nation having its own supply of iron ore, and ample metallurgical fuel of desirable quality, has an advantage in the race for industrial supremacy. The total quantity of pig iron produced throughout the world is in the neighborhood of 73,000,000 tons. This is exclusive of metal obtained from forges by crude methods in a number of countries. Upon the above basis the United States at present supplies 41 per cent., and the combined output of the European nations in conflict is 55 per cent. of the world's pig-iron product.

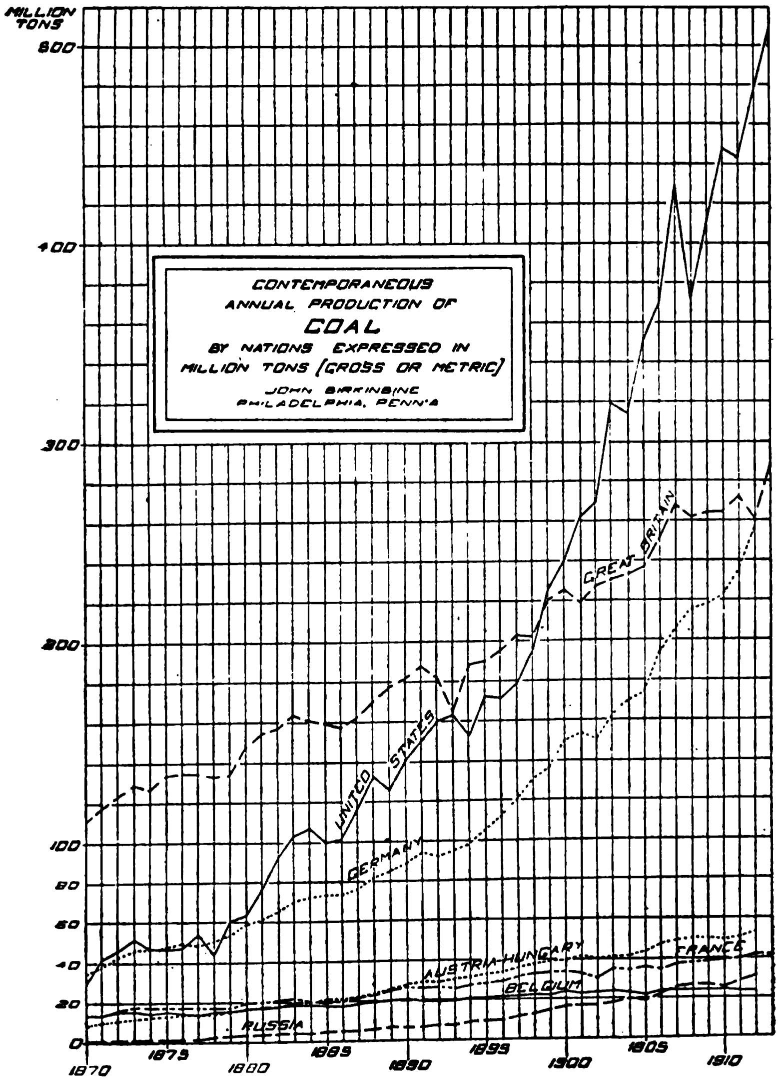


PLATE C.

Coal.—Plate C, which follows the form of Plates A and B, except that the spaces between horizontal lines represent 20,000,000 tons, illustrates the production of coal by various nations, and when read in connection with Plates A and B shows how different countries compare in the production and utilization of coal and iron ore.

England maintained preëminence in coal output until 1898, but since that date the United States has been in the lead, and in 1913 produced almost as much coal as its nearest competitors, Great Britain, Germany and Luxemburg, combined. The United States supplies more coal than all of continental Europe; in fact, as much as the balance of the world excluding Great Britain—39 per cent.

The coal product of the United States and Great Britain averages higher grade than that of continental Europe, where large quantities of lignite or brown coal are secured, but the statistics given include the production of anthracite, bituminous, and lignite or brown coal.

In Plate C only those countries contributing 20,000,000 long or metric tons annually are indicated, but other countries mine large quantities of coal. Japan, China, India and Canada each produces between 10,000,000 and 20,000,000 tons annually, and New South Wales, Spain, Transvaal, Natal, New Zealand, Mexico, Holland and Chile each supplies between 1,000,000 and 10,000,000 tons yearly. About thirty countries produce smaller amounts.

In round numbers the total annual production of coal in the world at the present time is 1,250,000,000 tons, of which the United States contributes between 38 and 39 per cent., and the combined output of the European nations now at war approximates 50 per cent.

As supplementary to the three charts, the following statements are presented to show the position of each nation as a producer of iron ore, pig iron and coal. To bring the statistics close to the present time, the obtainable records subsequent to the last census are given. The data are not completely contemporaneous, owing to the different periods for which information is available and to delays in making official statistics public.

The charts show that, according to latest official comparable data, the countries mentioned rank in volume of products in the following order:

Iron ore.	Pig iron.	Coal.
United States,	United States,	United States,
Germany,	Germany,	Great Britain,
France,	Great Britain,	Germany,
Great Britain,	France,	Austria-Hungary,
Spain,	Russia,	France,
Sweden,	Austria-Hungary,	Russia,
Russia.	Belgium.	Belgium.

Analyses of these statements and of the three diagrams present some interesting, if not surprising, features.

The relative importance of the nations as producers of coal, iron ore and pig iron is evident from the statistics presented, but the population of the countries and the relation which this bears to the quantities produced are of interest.

Thus, in per capita production, the nations rank:

As producers of iron ore.	As producers of pig iron.	As producers of coal.
Sweden,	United States,	Great Britain,
United States,	Belgium,	United States,
Germany,	Germany,	Germany,
France,	Great Britain,	Belgium,
Great Britain,	France,	France,
Austria-Hungary,	Sweden,	Austria-Hungary,
Russia.	Austria-Hungary,	Japan,
	Russia.	Russia.

Progress in iron ore and coal mining, and in the manufacture of pig iron, between the years 1870 and 1910, shows as follows for the various countries, the quantities being quoted as per 1,000 inhabitants:

In the interval of forty years the population of the United States increased from 38,500,000 to 92,000,000; its coal output rose from 765 to 4,869 tons; its iron ores mined from 100 to 620 tons, and its pig iron made from 43 to 297 tons per 1,000 inhabitants, the rate of increase being most pronounced in the last twenty years.

In the same period Great Britain's population advanced from 32,000,000 to 45,000,000, and its coal output increased from 3,685 to 5,994 tons, but its iron-ore production decreased from 513 to 342 tons (in the decade ending 1901 to 292 tons). The amount of pig iron manufactured rose from 208 to 214 tons per 1,000 inhabitants, reaching 231 tons in 1881.

In Germany and Luxemburg the population was augmented from 41,000,000 in 1871 to 66,000,000 in 1910; its coal output advanced from 922 to 3,360 tons, its iron ore production from 106 to 434 tons, and the pig iron manufactured from 38 to 224 tons per 1,000 inhabitants.

In forty years the population of France advanced from 36,000,000 to 39,600,000, and contemporaneously its proportionate coal production grew from 438 to 991 tons, its iron ore output from 85 to 404 tons, and pig iron manufactured from 34 to 113 tons per 1,000 inhabitants.

In Austria-Hungary the increase in population was from 36,000,000 in 1870 to 49,000,000 in 1910. In the same interval its coal output was augmented from 233 to 989 tons, its iron ore product from 32 to 92 tons, and the amount of pig iron manufactured from 13 to 41 tons per 1,000 inhabitants.

From 1870 to 1912 the population of Russia has risen from 85,000,000 to 171,000,000, and in that interval its coal mined advanced from 8 to 179 tons, its iron-ore output from 9 to 41 tons, and its pig iron manufactured from 4 to 24 tons per 1,000 inhabitants.

Belgium produces but little iron ore, but, while its population advanced from 5,000,000 in 1870 to 7,500,000 in 1910, its proportionate coal output rose from 2,692 to 3,222 tons, reaching 3,505 tons in the year 1900. This nation is a liberal manufacturer of pig iron, of which it produced 111 tons in 1870 and 249 tons in 1910 per 1,000 of population.

Sweden differs from Belgium in having but a small coal supply but an abundance of iron ore. In the forty years the population advanced from 4,000,000 to 5,500,000, its iron-ore output, much of which is exported, rose from 151 to 1,006 tons, and its pig-iron production using charcoal as fuel from 72 to 109 tons per 1,000 inhabitants.

To illustrate the position of various nations as producers of coal, iron ore and pig iron, a series of statistical statements has been prepared covering the interval from 1870 to the present time, divided by decades. These show for each nation the total production in a decade, also the mean, maximum and minimum annual productions.

Reference has been made to the interdependence of nations, and, without attempting detailed explanation, this is epitomized in statements from latest data available of the amounts of iron ore, coal and pig iron exported and imported by various countries.

In the *United States* all iron ore produced, except about 1,000,000 tons annually exported to Canada, is smelted in the country, and in addition, in late years, from 2,000,000 to 2,600,000 tons of iron ore are imported, principally from Cuba, Sweden and Newfoundland.

The imports of pig iron and ferro alloys in late years into the United States range from 130,000 to 237,000 tons, and the exports from 60,000 to 275,000 tons, showing that the domestic pig iron is used at home.

From 12,000,000 to 22,000,000 tons of coal are annually exported from and 1,250,000 to 2,000,000 tons imported into the United States.

Germany and Luxemburg smelt their domestic iron ores, and in addi-

tion import 11,000,000 to 14,000,000 tons, principally from Sweden, France and Spain, and have exported about 2,500,000 tons, chiefly to Belgium and to France.

Germany annually imports 125,000 to 140,000 tons, and exports in the neighborhood of 1,000,000 tons of pig iron.

Thirty-one million to thirty-four million five hundred thousand tons of German coal are exported to Austria-Hungary, Holland, Belgium, France, Sweden and Russia, and from 17,500,000 to 18,500,000 tons are imported into Germany.

Great Britain, in addition to its domestic iron-ore supply, draws 6,000,000 to nearly 8,000,000 tons from foreign mines, two-thirds coming from Spain, although Algeria, Norway, Sweden, Tunis, Greece and France contribute considerable quantities. No ore is exported.

Great Britain exports from 1,000,000 to 2,000,000 tons of pig iron, but the imports seldom exceed 250,000 tons.

Of its large fuel production, 65,000,000 tons of coal and about 1,000,000 tons of coke are exported to continental European, South American, and other countries. Those receiving more than 5,000,000 tons of British fuel annually are France, Italy, Germany, Sweden and Norway.

France.—Four million to over eight million tons of French iron ore are annually exported, principally to Belgium and Germany, and from 1,000,000 to 2,000,000 tons of iron ore are imported, mostly from Germany and Spain. France exported from 100,000 to 250,000 tons of pig iron annually, while its imports of this metal are less than half these amounts.

France imports (mostly from England) about 16,000,000 tons of coal and 2,000,000 to 2,750,000 tons of coke, while its fuel exports are from 1,300,000 to 2,000,000 tons.

Russia exports little iron ore, but about 4,000,000 tons of coal and 500,000 or more tons of coke are imported.

Austria-Hungary imports annually from 500,000 to 600,000 tons of iron ore, mainly from Sweden, and exports about 100,000 tons, principally to Germany. From 100,000 to 200,000 tons of pig iron are imported and 28,000 to 60,000 tons exported.

From 9,000,000 to 12,000,000 tons of coal are imported, principally from Germany, and 8,000,000 tons, mostly lignite, are exported, chiefly to Germany.

Spain is a large iron-ore producer, and exports 7,000,000 to 9,000,000 tons per year, the greater portion going to Great Britain. It produces 300,000 to 400,000 tons of pig iron, and its imports and exports are small. The domestic production of coal is augmented by importations of 2,000,000 tons annually.

Belgium mines from 125,000 to 250,000 tons of iron ore annually, but imports mostly from France, Germany and Spain 5,000,000 to 6,000,000 tons, about 500,000 tons of which are reëxported to Germany.

Belgium fabricates practically all of its pig-iron product, and also imports about two-thirds of a million tons annually, the larger part from Germany, although Great Britain and France also furnish pig iron. It mines from 22,000,000 to 24,000,000 tons of coal, imports 6,000,000 to 7,500,000 tons, and exports about 5,000,000 tons, principally to France.

Sweden produces an abundance of high-grade iron ore, the greater part of which is exported to Germany, England, Belgium and the United States. The pig-iron industry depends almost entirely on charcoal, about one-fourth of the production being exported.

The various nations have been discussed independently of any colonies or dependencies, for, as before stated, the pig iron made has not been followed through mills and factories. But the statements presented may indicate how intimate the association has been between nations now combating each other and seeking to cripple their adversaries in every possible manner. These may also suggest the present and prospective status of

each country as a contributor to the supply of resources necessary for the world's advancement, and show how changes of boundary lines may enlarge or reduce the independent manufacturing possibilities of the nations now at war.

On preceding pages reference has been made to the necessary interchange of the raw or the manufactured materials between the countries now at conflict, and it is possible that the result of the war may greatly strengthen some or seriously cripple others of the contestants, according to the peace terms which may be agreed upon. The commercial supremacy of each nation is, to a great extent, controlled by its ability to manufacture articles of iron and steel.

The effects of such changes may extend far beyond the limits of the countries contending with each other, and the possession of natural reserves of some nations or colonies and also in the Orient may be more than a mere incident in the European War.

PRODUCTION OF IRON ORE in the Eight Principal Iron-Ore Mining Countries in the World, from 1870 to 1913, by Ten-year Periods.

UNITED STATES—Long Tons.

Period.	Total production.	Mean annual production.	Year.	Maximum.	Year.	Minimum.
1870–79	44,645,440	4,464,544	1879	5,255,000	1876	3,734,000
1880–89	95,089,403	9,508,940	1889	14,518,041	1880	7,120,362
1890–99	163,989,193	16,398,919	1899	24,683,173	1893	11,587,629
1900–09	383,932,500	38,393,250	1907	51,720,619	1900	27,553,161
1910–13	218,022,042	54,505,510	1913	61,980,437	1911	43,876,552

GERMANY AND LUXEMBURG—Metric Tons.

1870–79	51,160,842	5,116,084	1873	6,177,576	1870	3,839,222
1880–89	89,499,306	8,949,931	1889	11,002,187	1880	7,238,640
1890–99	133,313,059	13,331,306	1899	17,989,635	1891	10,657,521
1900–09	224,434,495	22,443,450	1907	27,697,127	1901	16,570,182
1910–13	128,284,212	32,071,053	1913	35,941,285	1910	28,729,700

GREAT BRITAIN—Long Tons.

1870–79	156,173,886	15,617,389	1876	16,841,584	1870	14,370,655
1880–89	158,823,859	15,882,386	1882	18,031,957	1887	13,098,041
1890–99	130,184,239	13,018,424	1899	14,461,330	1893	11,203,476
1900–09	142,877,457	14,287,746	1907	15,731,604	1901	12,275,198
1910–13	60,533,158	15,133,290	1913	15,997,328	1912	13,790,391

FRANCE—Metric Tons.

1870–79	25,715,612	2,571,561	1872	3,081,026	1871	2,099,706
1880–89	28,743,748	2,874,375	1882	3,467,251	1886	2,285,648
1890–99	40,088,780	4,008,878	1899	4,985,702	1890	3,471,718
1900–09	76,317,045	7,631,705	1909	11,890,000	1902	5,003,782
1910–12	49,410,078	16,470,026	1912	18,800,000	1910	14,605,542

SPAIN—Metric Tons.

1870–79	9,419,302	941,930	1874	423,401	1877	1,576,150
1880–89	46,444,883	4,644,488	1887	6,796,266	1881	3,502,681
1890–99	63,292,106	6,329,211	1899	9,397,733	1892	5,041,317
1900–09	87,235,290	8,723,529	1907	9,896,178	1902	7,904,555
1910–11	17,440,486	8,720,243	1911	8,773,691	1910	8,666,795

RUSSIAN EMPIRE—Metric Tons.

Period.	Total production.	Mean annual production.	Year.	Maximum.	Year.	Minimum.
1870–79	9,166,008	916,601	1875	1,056,545	1871	794,779
1880–89	11,692,556	1,169,256	1889	1,646,840	1883	998,393
1890–99	30,468,011	3,046,801	1899	5,788,709	1890	1,802,165
1900–09	50,621,536	5,062,154	1900	6,122,270	1902	4,004,007
1910–11	12,750,218	6,375,109	1911	6,985,039	1910	5,765,179

SWEDEN—Metric Tons.

1870–79	7,466,982	746,698	1874	926,825	1870	630,739
1880–89	8,883,492	888,349	1889	985,904	1880	775,344
1890–99	17,402,164	1,740,216	1899	2,435,200	1890	941,241
1900–09	38,011,618	3,801,162	1908	4,713,160	1900	2,609,500
1910–13	25,886,414	6,471,604	1913	7,479,393	1910	5,552,678

AUSTRIA-HUNGARY—Metric Tons.

1871–79	10,246,331	1,138,481	1873	1,588,058	1877	884,260
1880–89	14,632,594	1,463,259	1889	1,780,772	1881	1,084,443
1890–99	25,162,976	2,516,298	1899	3,293,004	1892	1,913,831
1900–09	37,511,679	3,751,168	1908	4,568,814	1903	3,155,116
1910–11	9,245,998	4,622,999	1911	4,712,666	1910	4,533,332

PRODUCTION OF PIG IRON in the Eight Principal Pig-iron Producing Countries in the World, from 1870 to 1914, by Ten-year Periods.

UNITED STATES—Long Tons.

1870–79	21,885,266	2,188,527	1879	2,741,853	1870	1,665,179
1880–89	51,534,529	5,153,453	1889	7,603,642	1880	3,835,191
1890–99	93,538,215	9,353,822	1899	13,620,703	1894	6,657,388
1900–09	197,807,609	19,780,761	1909	25,795,471	1900	13,789,242
1910–14	134,978,776	26,995,755	1913	30,966,301	1914	23,332,244

GERMANY AND LUXEMBURG—Metric Tons.

1870–79	19,272,726	1,927,273	1873	2,240,575	1870	1,391,124
1880–89	36,195,906	3,619,591	1889	4,524,558	1880	2,729,038
1890–99	58,777,609	5,877,761	1899	8,143,132	1891	4,641,217
1900–09	105,500,005	10,550,001	1907	12,875,159	1901	7,880,087
1910–14	81,683,742	16,336,748	1913	19,309,172	1914	14,389,547

GREAT BRITAIN—Long Tons.

1870–79	63,796,993	6,379,699	1872	6,741,929	1870	5,963,515
1880–89	79,127,923	7,912,792	1882	8,586,680	1886	7,009,754
1890–99	79,614,624	7,961,462	1899	9,421,435	1892	6,709,255
1900–09	91,251,065	9,125,107	1906	10,149,388	1901	7,761,832
1910–13	39,306,601	9,826,650	1913	10,481,917	1912	8,889,124

FRANCE—Metric Tons.

1870–79	13,364,987	1,336,499	1878	1,521,274	1871	859,641
1880–89	17,723,834	1,772,383	1883	2,069,430	1886	1,516,574
1890–99	21,919,062	2,191,906	1899	2,567,388	1891	1,897,387
1900–09	30,278,382	3,027,838	1907	3,590,235	1901	2,388,823
1910–13	18,758,948	4,689,737	1913	5,311,316	1910	4,038,297

RUSSIA—Metric Tons.

Period.	Total production.	Mean annual production.	Year.	Maximum.	Year.	Minimum.
1870–79	3,805,983	380,598	1879	420,660	1871	339,513
1880–89	5,278,717	527,872	1889	733,720	1880	427,696
1890–99	15,113,846	1,511,385	1899	2,677,120	1890	905,460
1900–01	27,634,376	2,763,438	1904	2,962,283	1903	2,449,701
1910–12	10,830,846	3,610,282	1912	4,197,635	1910	3,040,047

AUSTRIA-HUNGARY—Metric Tons.

1870–79	4,423,447	442,345	1873	534,508	1877	387,630
1880–89	6,838,158	683,816	1889	855,822	1880	464,234
1890–99	11,431,258	1,143,126	1899	1,467,664	1891	921,846
1900–09	16,119,729	1,611,973	1909	1,995,511	1904	1,375.865
1910–12	6,545,212	2,181,737	1912	2,372,660	1910	2,060,975

BELGIUM—Metric Tons.

1870–79	5,379,203	537,920	1872	655,565	1879	389,330
1880–89	7,323,421	732,342	1889	832,226	1880	608,084
1890–99	8,631,404	863,140	1899	1,036,185	1891	684,126
1900–09	12,335,766	1,233,577	1909	1,616,370	1901	764,180
1910–13	8,726,830	2,181,708	1913	2,527,070	1910	1,852,090

SWEDEN—Metric Tons.

1870–79	3,343,027	334,303	1876	352,467	1871	298,761
1880–89	4,329,398	432,940	1885	464,737	1882	398,945
1890–99	4,873,948	487,395	1897	538,197	1893	453,421
1900–09	5,401,295	540,130	1907	615,778	1909	444,764
1910–13	2,668,404	667,101	1913	730,257	1910	603,939

PRODUCTION OF COAL in the Eight Principal Coal-Mining Countries in the World, from 1870 to 1913, by Ten-year Periods.

UNITED STATES—Long Tons.

1870–79	476,566,017	47,656,602	1879	60,808,749	1870	29,496,054
1880–89	1,019,753,264	101,975,326	1888	132,701,827	1880	63,822,830
1890–99	1,712,331,738	171,233,174	1899	226,554,635	1890	140,866,931
1900–09	3,337,185,589	333,718,559	1907	428,895,914	1900	240,789,310
1910–13	1,877,082,366	469,270,592	1913	508,971,540	1911	443,054,614

GREAT BRITAIN—Long Tons.

1870–79	1,274,494,850	127,449,485	1877	134,179,968	1870	110,431,192
1880–89	1,607,990,447	160,799,045	1889	176,916,724	1880	146,969,409
1890–99	1,910,785,455	191,078,546	1899	220,094,781	1893	164,325,795
1900–09	2,414,416,661	241,441,666	1907	267,830,962	1901	219,046,945
1910–13	1,084,171,738	271,042,935	1913	287,430,473	1912	260,416,338

GERMANY—Metric Tons.

1870–79	456,561,932	45,656,193	1879	53,470,716	1870	34,003,004
1880–89	719,117,229	71,911,723	1889	84,973,230	1880	59,118,035
1890–99	1,073,735,837	107,373,584	1899	135,824,427	1890	89,051,527
1900–09	1,791,128,249	179,112,825	1909	217,445,656	1900	149,788,256
1910–12	711,706,424	237,235,475	1912	255,810,094	1910	222,375,076

FRANCE—Metric Tons.

Period.	Total production.	Mean annual production.	Year.	Maximum.	Year.	Minimum.
1870–79	161,713,709	16,171,371	1873	17,479,341	1871	13,258,920
1880–89	208,703,065	20,870,307	1889	24,303,509	1881	19,361,564
1890–99	286,581,124	28,658,112	1899	32,863,000	1893	25,650,981
1900–09	346,903,640	34,690,364	1909	37,840,086	1902	29,997,470
1910–13	159,716,914	39,929,229	1912	41,145,178	1910	38,349,942
AUSTRIA-HUNGARY—Metric Tons.						
1870–79	121,822,412	12,182,241	1879	14,891,024	1870	8,355,944
1880–89	202,430,110	20,243,011	1889	25,328,417	1880	16,128,718
1890–99	325,647,572	32,564,757	1899	38,739,000	1890	27,504,032
1900–09	439,719,257	43,971,926	1908	49,626,184	1900	39,502,301
1910–12	151,031,218	50,343,739	1912	52,521,776	1910	48,649,768
RUSSIA—Metric Tons.						
1870–79	15,555,667	1,555,567	1879	2,874,790	1870	696,673
1880–89	43,237,790	4,323,779	1889	6,213,869	1880	3,286,534
1890–99	91,966,854	9,196,685	1899	14,311,200	1890	6,085,080
1900–09	204,227,251	20,422,725	1909	26,075,086	1900	16,135,600
1910–12	83,551,747	27,850,582	1912	30,646,163	1910	24,898,345
BELGIUM—Metric Tons.						
1870–79	146,893,125	14,689,313	1873	15,778,401	1877	13,669,077
1880–89	179,771,122	17,977,112	1889	19,869,980	1881	16,873,951
1890–99	206,800,233	20,680,023	1898	22,075,093	1893	19,410,519
1900–09	231,237,587	23,123,759	1907	23,705,190	1905	21,775,280
1910–12	69,942,240	23,314,060	1910	23,916,560	1912	22,972,140
JAPAN—Metric Tons.						
1881–89	12,808,235	1,423,136	1889	2,388,614	1881	925,198
1890–99	45,721,857	4,572,186	1898	6,761,301	1890	2,608,284
1900–09	115,283,197	11,528,320	1909	14,973,617	1900	7,429,457
1910–12	52,953,789	17,651,263	1912	19,639,755	1910	15,681,324

—"Journal of the Franklin Institute."

TRIPLE AND QUADRUPLE SCREWS.

Twin, triple and quadruple screws are at present employed for the propulsion of large battleships. In the United States Navy there are several Dreadnaughts with twin screws driven by reciprocating machinery, whilst several large German and French warships are driven by three screws. The British practice is to have quadruple screws. Several considerations are taken into account when the number of screws are decided upon, such as the total power to be distributed. When this is not excessive either triple or quadruple may be adopted, and the question of the propulsive efficiency of these systems is then, amongst other things, taken into consideration. From experiments made in the Spezia tank it appears that higher efficiency may be looked for from a triple arrangement than from quadruple screws. Four screws have several disadvantages, amongst them being an increase in resistance due to the appendages of the shafts and a smaller utilization of the wake. In a triple-screw ship the center propeller is placed most advantageously for making use of the wake, whereas in quadruple screws the wing propellers are placed far out from the center line and are thus in a position of inferiority. It has also been shown by experiment that considerable interference is effected by the propeller race

from the forward pair of screws upon the wake of the after pair in a quadruple arrangement. In triple-screw warships driven by turbine machinery the distribution of power is equal and the three screws are generally of the same dimensions. In triple-screw merchant ships, fitted with the combination type of machinery, two large screws are placed on the wings and a small screw, driven by the turbine, is placed in the center position. It is most important from the point of view of propulsive efficiency that too much power should not be passed through the small center screw, as although the wake value is high, the propeller is generally too small to use the power efficiently. Although experiments indicate that triple screws are more efficient than quadruple, high propulsive efficiencies have been obtained with both arrangements, the triple-screw light cruisers being very efficient, as also were the quadruple-screw battle cruisers of the *Invincible* type.—"Shipbuilding and Shipping Record."

MODERN SUBMARINE TORPEDO BOATS OF THE UNITED STATES AND OTHER NAVIES.

By Herbert S. Howard, Naval Constructor, U. S. N.

Submarines in the present war have come so much to the front that not only navy men but the laymen throughout the country are evincing great interest in them. The questions which most frequently come to the front seem to be: How do they submerge? How deep do they go? How do they see? How fast can they go? How is it that German submarines can go away over into the Irish Sea or other waters about England and return to Germany? Do they carry guns as well as torpedo tubes? And very often: Are our submarines as good as the German submarines we read of?

In the first place, submarine boats must be strongly built, as they must withstand the pressure of water when they are submerged. For this reason the structure of submarines is heavy when compared to surface vessels, and must be made of a shape capable of resisting high pressure. It is the practice in all countries to design submarines to withstand submergence to depths of 150 to 200 feet, and beyond such depths they must not go.

To submerge a submarine boat water is admitted to ballast tanks, thus destroying the reserve buoyancy of the vessel. When it is desired to bring the vessel to the surface, water is pumped or blown by compressed air from these tanks. In some types of submarines these tanks are located within the main hull, which is circular in section. In some other types the tanks are located outside of the circular-section hull, between it and a light exterior hull, more or less ship-shaped. Our submarines and those of England are practically all of the first type, while those of Germany and France are of the latter type. In all submarines, however, water is taken in to submerge.

When submerged, submarines ordinarily receive no fresh air, and a submarine may remain submerged for about 10 hours before the air becomes too foul for breathing. In many foreign submarines devices for regenerating the air are fitted, to permit them to remain submerged a much longer time, and it is worth noting that devices of this nature have been tested and are now under consideration for the submarines of our Navy. When running submerged the submarine is supplied with one or more periscopes, with which it can see, provided the vessel is only a short distance below the surface. These are optical instruments with lens systems similar to those of a telescope, but with prisms at the top and bottom to turn the rays of light down, and out at the bottom. When completely

submerged, however, the submarine is blind. Various forms of searchlights, etc., have been suggested for under-water vision, but so far none have promised success.

Speed in submarines is a feature much sought after. Due to limitations of electric power, size of battery, etc., the submerged speed cannot at present be brought to a high figure. In modern submarines this varies from 9 to 11 knots, and this speed can be maintained for only one hour. If a slower speed is used the distance covered will be greater, but the maximum submerged radius of a modern submarine will probably be 100 miles at 5 knots.

Surface speed offers greater possibilities of development and increase, and along these lines all countries are now working. For several years 14 knots has been considered about a maximum, but with development in oil engines, greater speeds have been attained. Some of the English submarines in service make 16 knots, and some German submarines 17 or 18 knots. At the present time, however, both in this country and abroad, there are submarines building of 20 and 21 knots speed. In such vessels steam or Diesel engines are used. None of these fast vessels are in service, so that it remains for the future to show whether they will usurp the place of the torpedo-boat destroyer. The radius of action of a modern submarine on the surface varies from 2,500 to 5,000 miles, so it may be seen that long trips are possible.

All modern submarines carry torpedoes as their essential armament. In general, it is foreign practice to fit two or more tubes in the bow and two in the stern, while in our Navy we have adhered to a bow arrangement with two to four tubes grouped there together. To fire these torpedoes the periscope must, of course, be exposed and the vessel herself pointed at the enemy. It is not always realized, and so may be worth noting, that the torpedo itself is really a small submarine boat with high-powered engines. With modern torpedoes speeds of from 35 to 40 knots are obtained, with ranges up to 4,000 yards or more. Submarines, however, usually fire their torpedoes at ranges of from 600 to 2,000 yards. In addition to torpedoes submarines nowadays generally carry one or two small guns, and the use to which such guns may be put has been brought out clearly in the present war.

It is often thought that a submarine is a mass of machinery and that the life on board is simply existence in a whirl of shafts, levers, etc. A submarine is indeed filled with machinery and the crew are mechanical experts; but the quarters provided for the crew are as good as those on a torpedo boat. Bunks are furnished for all the men; electric ranges are installed for cooking food, and everything is done to make these vessels habitable and comfortable as far as possible for comparatively long cruises. For example, fuel, supplies, fresh water, etc., can be carried by modern submarines to make a cruise of 5,000 miles. This is a fact not generally realized.

A modern submarine—and this applies to our own as well as other navies—is a vessel displacing about 450 tons on the surface and 600 tons submerged. The surface speed is from 13 to 15 knots and the submerged speed from 10 to 11 knots. Such a vessel would be 150 to 160 feet long. Within the boat a certain space is taken by ballast tanks, for, as I have stated, water is admitted to destroy the buoyancy and to permit the vessel to submerge. The remainder of the space is given up to engines, electric storage battery and motors, torpedoes, crew's quarters and central operating compartment. In this last-named space are grouped the important means for controlling the vessel—the controls for flooding ballast tanks, the air connections for blowing water out of them, the wheels for controlling the steering rudders and the diving rudders, and the periscopes.

When the boat is to submerge all the deck gear is taken down and all openings in the hull are tightly closed before water is admitted to the

tanks. When the boat is running on the surface of the water rail stanchions are fitted, a small bridge is rigged, radio masts are hoisted, and the submarine becomes a small cruising vessel.

In our Navy we have kept pace with the world in submarine development. Most of our submarines are of about the size and type described in the foregoing; but in addition we are building at present one large submarine of about 1,000 tons surface displacement and of about 20 knots speed, and appropriations for two more such vessels were recently made by Congress. This program parallels those of England, France and Germany, which, in addition to their submarines of 400 or 500 tons surface displacement, are also building submarines of from 1,000 to 1,100 tons and of from 20 to 21 knots speed.—"Engineering News."

GEARED MARINE STEAM TURBINES.

Mr. J. Hamilton Gibson, Wh. Ex., M. I. N. A., engineering manager of Messrs. Cammell, Laird & Co., Birkenhead, read a very interesting paper before the members of the Foremen's Mutual Benefit Society of the Birkenhead and Liverpool district. Mr. Roy M. Laird presided.

Mr. Gibson said the steam turbine, excellent as it is in many respects, especially for large powers and high-speed vessels, has certain well-defined limitations. It is a grand motor at full speed; but its efficiency falls off very rapidly if for any reason it is slowed down. Then it becomes what is popularly known as a "steam-eater." When once the stop valve is opened the steam rushes through the turbine at a speed averaging 1,000 feet per second, or 700 miles per hour—a hurricane is a gentle breeze in comparison. And if the turbine is to make the best use of this steam, it must revolve at such a speed that the circumferential velocity of the blades is not less than half the steam speed. No use closing down the throttle, the steam will not move an inch slower on that account; in fact, it gets through the turbine all the faster, because there is less of it to crowd through. To slow down a reciprocating engine, you merely bite off small mouthfuls of steam, imprison it in successive closed cylinders, and expand it just as slowly and deliberately as the pistons will permit. But there are no closed spaces or pistons in a turbine, once the stop valve is opened there is a clear run to the condenser through myriads of passages, and the steam will fly through at a speed fixed by the immutable laws of nature, whether it is an ounce or a ton. To increase the power and reduce the speed we must not only provide a sufficient number of openings in the first ring of blades, but we must also provide a large number of successive rings of blades, increasing in size to allow for expansion, and in order to extract the last ounce of energy from the steam particles before we allow them to escape. It means increasing the size and, therefore, the weight of the turbine, and that cannot always be conveniently done. In short, we sometimes reach a point where, if the turbines are to revolve slow enough for practical use they would have to be made so big and heavy that they would sink the ship. On the other hand, if it is decided to keep down the size of the turbines, and run them faster, the propeller would have to be made so small, and would revolve at such a tremendous "lick," that it would merely tear a hole in the water—the blades would lose their grip—and the propulsive effect would be practically *nil*. For vessels of high speed, 20 or 30 knots or over, the turbine may be connected direct to the propeller shaft, and results can be attained equal or superior to the best reciprocating-engine practice. In fact, when we get beyond speeds of 25 knots, which is about the practical limit for reciprocating engines, the turbine rapidly overhauls the reciprocator in all round

efficiency, and has driven it completely off the field, as in all modern war vessels, where high speed is a paramount consideration.

But if we take the merchant steamship tonnage of the world, it will be found that only about 2 per cent. of it comprises vessels of over 20 knots. The remaining 98 per cent., including ocean-going tramps and intermediate liners, are still "hacking through," most of them at 10 to 12 knots, by means of the good old piston engines. The limit of economy has long ago been reached in these engines by quadruple and even quintuple expansion, to say nothing of super-heaters, feed heaters, and patent packings. It was, therefore, high time that the marine engineering faculty turned its attention to the question as to how the steam turbine, which had so revolutionized the performances of high-speed vessels, could be adapted for use in this vast fleet of steamers, which comes below the 20-knot line. This includes some 1,500 new vessels built annually, totalling about 3,000,000 tons, so that the matter is well worth consideration. Obviously some form of gearing would be required to link the naturally fast-running turbine to the large, efficient, slow-turning propeller. In view of the large powers to be transmitted, great importance was attached to the strength of the gear wheels, the size of the teeth, and various methods of carrying the gearing in shock-absorbing frames, or even providing the wheels with built-up or laminated flexible teeth. When gearing was first proposed, old engineers shook their venerable heads in fear and doubt as they recalled the geared engines of half a century ago, and remembered the numerous troubles and breakdowns of those early days. But here was quite a different problem. The first marine engines were ponderous and slow, making only 10 or 20 revolutions per minute, and had to be geared up to suit the propeller. The steam turbine, running at 2,000 or 3,000 revolutions per minute, requires to be geared down, and that makes all the difference. Moreover, the old gear wheels had to be built up with large wooden teeth morticed into the rim, carefully shaped and trimmed and fitted by hand. Today we have machine tools of the utmost precision that not only cut, but actually generate the theoretically correct form of the teeth to within a thousandth of an inch error. The original inventor and perfector of the marine steam turbine took little or no part in the paper battles of the gearing. Sir Charles Parsons is not the man to advertise his ideas "for daws to peck at" before he has demonstrated the facts to his own satisfaction. How did he proceed? Whilst the controversy was raging he quietly bought an old tramp steamer, removed the triple-expansion engines, and fitted in their place a set of geared turbines. Nothing else was altered—same boilers, same shafting, same old propeller even, running at about 70 r.p.m. and giving a speed of about 9 knots. A typical tramp. A toothed wheel, 8 feet 4 inches in diameter, was fitted in the original shaft, and gearing into this wheel were two pinions 5 inches diameter, that on the starboard side being driven by a high-pressure turbine and the port pinion by the low-pressure turbine. The turbines ran at about 1,400 revolutions per minute, the gear ratio being thus 20 to 1. Here it will be well to compare the performance of this vessel—the *Vespasian*—with the results obtained with her original triple-expansion reciprocating engines: (1) Twenty-five tons' weight of machinery was saved; (2) under normal full-speed steaming conditions an increased speed of 1 knot was obtained, with the same coal consumption; (3) at 65 revolutions per minute the saving in coal amounted to 15 per cent., and at full speed, or 70 revolutions per minute, the saving was 19 per cent.; (4) this means that for the same power only five-sixths of the boiler power is needed, or, in other words, one boiler out of six could be saved—no small achievement. How does it come about that we get the same power and energy out of the imprisoned demon in a small enclosed cylinder as we used to get from a mighty engine with its great cranks, connecting rods and pistons? It's all a matter of speed!

The great point to keep in mind is this—and the late Robert Humphries used to lay great stress on it whenever he was asked to add any complication to his engines, which, in his opinion, reduced their simplicity without adequate return—that the main and sole object of any engine, however big and powerful, is merely to revolve a certain shaft and turn a propeller so many times a minute, and that, other things being equal, we get the same horsepower from a 3-inch shaft revolving 3,000 times a minute as we do from a 12-inch shaft making 50 turns a minute. Now, the essence of the design of a geared-turbine installation may be stated thus: A vessel of a given size and speed is best propelled by a screw of a certain diameter revolving at a certain number of revolutions per minute. Then, according to the weight and space that can be allowed for the machinery, we can design a set of turbines that shall revolve six times or twenty times as fast as the propeller, and introduce pinions and gear wheels to correspond. The beauty of the arrangement is that we are free to choose the best speed of rotation for the propeller, and at the same time the most efficient speed of rotation for the turbines. Another great advantage is this, that the speed of the ship can be reduced considerably if required, without adversely affecting the efficiency of the turbines, which, of course, means economy in fuel consumption. These turbines are connected direct to the propeller shaft, and must perforce revolve at the same speed; they never, or hardly ever, revolve fast enough to give the best economical results, and their efficiency falls off rapidly as the vessel reduces speed. The first thing that strikes one in the geared turbine is the remarkably small pitch of the teeth. The majority of engineers imagined, and naturally so, that enormously strong and heavy teeth would be required to transmit the enormous powers in modern ocean liners, amounting to some 15,000 H.P. per shaft in some cases. The normal standard pitch for all large powers is only .583 inch (less than ⅝ inch), which means that the teeth are less than 5/16 inch thick. This fine pitch, as we have seen, conduces to smooth running; but it is imperative that the wheel and pinion bearings be mostly rigidly held at the absolute right centers, and the standard of workmanship must be the highest attainable. The correct geometrical form of tooth has received very careful consideration, and the standard form for all gears is now an involute (as opposed to the cycloidal shape), with a short addendum and large radii in the roots to conserve the strength of the teeth. The involute form enables all gears to be cut with the same standard hob, and ensures that all wheels and pinions, of whatever size, shall gear together correctly. Moreover, if it is desired to vary the oil clearance between the teeth by opening or closing the wheel centers a few thousandths of an inch, this can be done without affecting the efficiency of the gears in the slightest degree. It may be taken as an axiom that if the gear wheels and pinions are absolutely concentric, are cut absolutely true, are fitted with absolute correctness both as regards centers and parallelism, and are, of course, properly lubricated, the gears will run at all speeds and powers in dead silence. The earlier gears were admittedly imperfect, and, therefore, the noise was distinctly in evidence, sometimes rising to a continuous droning note, particularly if the teeth were fitted too close, with little or no oil clearance. When a turbine is connected directly to the propeller shafting no thrust block is required, as the steam, pressing aft on the turbine blades, is balanced by the propeller thrust pressing forward. But a geared turbine is not connected direct to the propeller shaft. Its spindle terminates in a claw-coupling, which engages a similar claw-coupling on the pinion shaft. The geared turbine must, therefore, be independently balanced, just like a turbo-dynamo, and this is accomplished in the usual way by balance cylinders on the rotor, all self-contained within the turbine casing. We must provide separately for the propeller thrust, by fitting the usual thrust block secured firmly to the ship's keelsons.

Thrust blocks of the sector type were fitted in our two-geared turbine vessels recently completed for the passenger service between Buenos Ayres and Monte Video. When the first boat went on trial there were not wanting Jeremiahs who foretold dire disaster. The idea of attempting to run vessels of 5,000 H.P. on a single thrust collar. But, for once, the prophets were mistaken, the end justified the means, and these blocks ran from the very first without the slightest trouble. Only the other day I saw a report from the engineer of one of the boats on her voyage to the River Plate. He said, "Thrust bearings ran quite cool, with the chill not off." I think it may be said that all future geared-turbine jobs will necessarily be fitted with the "Michel" thrust, and I see no reason why it should not apply equally well for reciprocating engines now that the principle is established; but here again it will take time, probably years, to persuade superintending engineers that the old-fashioned multi-collared thrust block has at last been superseded by something better, and that the "Michel" thrust, like the geared turbine, has come to stay. It has taken shipowners a long time to make up their minds to adopt geared turbines in long-distance ocean-going liners. The pioneer vessel *Vespasian* ran her demonstration voyages from the Tyne to Rotterdam as far back as 1909, but it was less than two years ago when the first order for a geared-turbine ocean liner was placed. Now, most of the big companies are having geared vessels built, and already the Cunard Company's *Transylvania* has made one or two eminently successful voyages. There is always a tendency among shipowners to "wait and see," and when they see that their rivals are able to run vessels as big as their own on 15 to 20 per cent. less coal, with a smaller engine and stokehold complement, and can carry more cargo, they will not be slow to follow suit. Moreover, as compared with a triple-expansion job, the weight of machinery is 10 to 12 per cent. less, whilst the saving in space occupied is about 25 per cent. It may be considered by some very unsportsmanlike, but the geared turbine engine has set itself resolutely against the practice of "racing." It doesn't matter how light the ship, or how rough the sea, whether the propeller is revolving in the water or whisking in mid-air as the vessel throws her stern up, there is no appreciable variation in the revolutions, and the life of the tail shaft and the comfort of all on board is increased in consequence. We can therefore predict with confidence a boom in geared-turbine steamers for ocean traffic before we are very much older; and it behooves all enterprising shipowners and shipbuilders to be in the swim.—"The Steamship."

CLEANING OF CONDENSER TUBES.

Zeitschrift f. d. gesamte Turbinenwesen, No. 15, 1915.

The municipal electric works at Bielefeld, Germany, made an experiment to clean condenser tubes of lime deposits by means of hydrochloric acid. Mixtures varying in strength from 1 to 2 to 1 to 10 were used for the metals under test, *i. e.*, brass, tin, copper, wrought iron and cast iron. The action of the diluted acid on the bright surfaces of the parts to be cleaned was small, excepting cast iron, which produced considerable quantities of hydrogen gas. Therefore, it is recommended to paint the cast-iron water chambers to protect them against the acid. By further trials the quantity of acid solution of 1 to 5 was ascertained that was needed to clean a single tube. To keep the diluted acid in constant circulation in the horizontal type of condenser built by M. A. N., the suction pipe of a small centrifugal pump was connected to the inlet branch of the condenser. The discharge pipe of pump was connected to

a cover plate of the condenser, to which was also connected a vent pipe which terminated under the surface of a water tank. The condensers in question contained about 258 cubic feet water. They were charged for 3 hours each with 3,792 pounds of hydrochloric acid of 22 degrees Baumé, and subjected for another 20 hours to the action of the acid contained in the remaining solution until the formation of gas at the vent pipe had practically ceased. The fine mud in the tubes was washed out by a strong stream of water which discharged into the sewer. By analysis it was found that the deposit contained 83 per cent. of carbonated lime with small additions of organic and inorganic substances. The cost of the acid for dissolving the hard lime coating in this case for each condenser amounted to .2918 cent per square foot cooling surface. The acid cost is based on a price of 51.64 cents per hundred. The cooling surface was about 7,000 square feet.

THE PURCHASE OF COAL ON SPECIFICATIONS.

At an informal meeting of the committee appointed by the American Society for Testing Materials, held in New York in May, 1914, in connection with a proposed form of "Specification and Contract for the Purchase of Coal for Use in Steam Power Plants," the following questions were discussed:

1. Ash content and B.t.u. value have been and are widely used as factors for establishing the quality of coal bought under specifications, but the general conclusion has been reached that regular B.t.u. determinations on every sample are unnecessary where all the coal comes from the same region and bed. In such case the B.t.u. value varies proportionately with the ash content, which, therefore, becomes a measure of heating value. When coal comes from different beds or regions, then B.t.u. values and analyses are of value.

2. Very few specifications in use make any mention of the clinkering qualities of coal, although this matter is of great importance and should be covered in all specifications. The fusing temperature of ash is used to measure this, but its method of determination should be further investigated before definite recommendations can be made.

3. Moisture specifications were also discussed. Neither the "as received" nor the "dry" basis has proved altogether satisfactory. Moisture in coal is variable; its exact determination in any shipment is difficult, because of uncontrollable losses, and its amount is largely accidental. To use the "dry-coal" basis may give inaccurate values when coals of different nature are compared. In fact, a satisfactory way of handling the moisture question has not yet been evolved.

4. Alternate methods of analysis are not desirable; a standard specification should always specify one standard method. The question of weights was also discussed, but no conclusions were arrived at.

As a general conclusion, it is believed by many chemists, engineers and others, that after the buyer of steam coal has decided on the description of coal he desires and what is best adapted to his use, there are only two items necessary, as regards analysis, in making up specifications, that is, the percentage of ash and the fusing temperature of the ash.

A standard percentage of ash of a given kind of coal (field or seam) insures a standard B.t.u. value; any change in ash percentage makes a corresponding change in B.t.u. value. The right ash-fusing temperature means no clinkers, and good combustion results if the coal is handled properly.

Where smoke laws are to be observed it may be necessary to specify also a maximum volatile content of the coal, although even here it would

generally be advisable and economical in the end to alter furnace and boiler conditions so that higher volatile, cheaper (usually) coals could be used.

Simplicity, equity and practicability should be the characteristics of all specifications for the purchase of coals.—F. R. Wadleigh, in "Coal Age."

GOOD AND BAD LUBRICANTS.

By G. Basil Barham, A. M. I. E. E.

Whatever an economist might advise or do, no engineer would select a lubricant on account of its lasting qualities alone. To him they would be quite a secondary consideration; he would require to be assured that it would keep cool the moving parts on which it was to be used before he would go into the question of whether it would or would not last longer than some other lubricant he was using or had under consideration.

The first essential of a lubricant, from an engineering point of view, is that it should possess a low and constant coefficient of friction. If the friction is small the heat generated will be small. Providing the lubricant used will float the parts and allow a film to be maintained, it only remains that the characteristics which tend to set up friction by resisting the relative movement of the parts, should be as few and as inappreciable as possible.

It is not easy, however, with all lubricants to maintain this film; with some it is not an easy matter to ensure an adequate supply passing to the point where it is required. Some lubricants will squeeze away and leave parts starved; others, of the composite variety may give up one or more of their constituents and leave the "filling" to clog and bind, and ultimately cut the surface of the parts. Further, if it were only a matter of a low coefficient of friction, practically any clean oil, whether animal, vegetable or mineral, would answer the purpose, provided it was fresh and in good condition. But any attempt to lubricate machinery for any length of time with either animal or vegetable oils would have disastrous results, as all organic oils undergo chemical changes, and either turn rancid, or thicken and gum. The only oil that can be used for lubricating purposes with any good and sustained result is a mineral oil, and unless this has been chosen, and treated specially by processes dependent upon expert knowledge not only of oils, but also of the purposes for which each is intended, the results are not likely to be satisfactory. It is emphatically not a case of "any old oil will do," where high-speed electrical and other machinery is in question; it is a question of comparative expense and expediency. Engines and generators cost more than lubricants, and it must be pointed out that oil, no matter how excellently it be blended, has its limitations, and there is a question as to whether, in view of modern developments, it is not bad engineering to use oil at all for lubricating purposes.

GREASES AND NON-FLUID OILS.

It is a fact that whilst types of prime movers have been standardized, and dimensions and proportions resulting from wide experience have been adopted for prime movers, one feature of design, *i. e.*, engine lubrication, has not kept pace with other developments. It is not too much to say that quite 80 per cent. of engine troubles are due to defective or improper lubrication. The other twenty per cent. can be about equally divided between careless or ignorant operation and inadequate proportions of work-

ing parts. At one time it was the practice to use oil lavishly in the cylinders of vertical condensing engines and with slide valves of the piston type, with the result that considerable damage was done to the boilers, and as soon as the bad effects of such oiling were made apparent a certain school of engineers straightway turned to the other extreme, and adopted the plan of running these parts without any oil or grease, trusting to the lubricating effect of the steam. Even then it is necessary to use oil on the flat slide valves of intermediate and low-pressure cylinders, so that the method is really of little or no use in preventing the access of oil to the feed water. When it is remembered that the globules of oil are so minute, much of the oil having been carried over in the form of vapor, that water only moderately emulsified can be allowed to stand in settling tanks for months and skimmed at regular intervals, without the whole of the oil content rising to the surface, it will be seen how difficult it is to deal with the trouble by means of filters of loofaa, straw, Hessian, or a combination of any of these with other substances. Chemical grease extractors have been introduced, some of which have been more or less successful, but as a general thing the difficulty has been so great that many power users have adopted the use of solid lubricants, and hold the opinion, to which many are being converted, that in some form these will be the lubricants of the future.

It is claimed that they increase the efficiency of oiling and minimize waste, and there is no doubt that, if of high grade, soap-solidified oils and greases can be successfully used on a number of parts which have to stand heavy duty. At the same time objections can be advanced against both these lubricants themselves and the methods in which they are generally used.

In the majority of cases solid or semi-solid lubricants are made by working up animal fats or vegetable butters with soap and water, or by thickening mineral oils of thin body but extreme viscocity with soap. The soaps used are of the ordinary lime, soda or lead types, and it must be pointed out that it is not an easy matter to prevent an excess of caustic soda being present. During use this is being converted into carbonate of soda, and there is a danger of the bearings being cut in consequence. Such greases frequently contain free fatty acids which attack the metal parts on which they are used; one of the least troubles experienced with such a lubricant would be the scoring of metal parts passing through stuffing boxes, as rods or valve stems, and also of the glands. Further, when thickened mineral oils are used, there is a liability for the oil content to be squeezed away, leaving the soap body behind it in the bearing, where it not only hardens but forms a nucleus to which any abraded metal particles and dirt adhere. In this way a gummy mass is formed which soon coats the moving parts, increasing friction and leading to ultimate cutting. Where such lubricants are used in boxes it is common practice to fit these with brass pins which rest in the part in motion. Should friction increase beyond the normal, and heating be set up, the theory is that these pins become heated also and cause the grease round them to flow to the journal. Should a bearing lubricated in this way run very hot through neglect, and escape notice, in a very short time the whole of the grease will have run through, and thus allow the brass pins to work down between the shaft and the brass or the cap of the bearing and so grind up, cutting both shaft and bearing.

THE ADVANTAGES OF NON-FLUID OILS.

The same objections cannot, however, be urged against high-grade non-fluid oils. They have all the advantages of soap lubricants but none of their disadvantages, and in addition they possess the property of adhering to metal surfaces with a force which is considerably above that possible

with fluid oils. They are all of high viscosity and can only be fed drop by drop. In that they prove economical, as with ordinary oil a greaser can and frequently does squirt oil with a lavish liberality not only on the part needing it, but also on any parts adjacent. It is not to be supposed that this slow feeding is a disadvantage even where a bearing has run hot, as, owing to its clinging properties, it films immediately on reaching the moving part and so forms an ideal lubricant. Non-fluid oils, also, will not gum, and they can be used without risk of clogging in central distribution systems provided the oil tubes are of sufficiently large internal diameter to permit of the oil flowing.

In practice it has been found that one pound of non-fluid oil of high quality is equal in lubricating effect to one gallon of high-grade fluid oil, and according to the average prices ruling the saving effected in direct cost by using such lubricants would be about 65 per cent., or more, according to the quality used.

The extremely low average coefficient of friction possessed by high-grade non-fluid oils has been proved by numbers of independent tests, and such oils are being largely used with high-speed machinery, being mechanically fed to the various parts. But where the adoption of non-fluid oils is contemplated, it must always be remembered that a lubricant which would suit a certain class of machinery might be of little use for another. As has been said, the art of economically employing a lubricant consists mainly in the application of precisely that quality of unguent which is best adapted to that particular place on each machine or each part of a machine on which pressure of lubricated surfaces of widely differing amount is found, and, above all, applying it in the best possible way.

THE USE OF GRAPHITE.

Of recent years the claims of graphite to be considered the ideal lubricant have been supported by a large and increasing number of engineers. It is common knowledge that when used for such purpose graphite should be of such quality and fineness as to give a perfectly smooth and even veneer or coating to the parts. The danger is that an impure or carelessly prepared grade might be used, which would carry into the bearings or cylinders such impurities as are naturally associated with graphite, as sand, clay, mica or talc. The essentials of good lubricating graphite are that it should be amorphous and of extreme softness. It should have neither luster nor brilliancy, and a heaped quantity of it should not reflect light, nor show bright particles when examined with a powerful lamp. The color is not so important; dense blackness does not necessarily show purity any more than a feeling of smoothness when rubbing a sample between the thumb and finger proves fineness. For example, it may be mentioned that by a special electrical process graphite can be made of such a degree of sub-division that the particles cannot be seen under an ordinary microscope. Ordinary graphite contains particles of about 1/350 of an inch in diameter, but electrically-prepared graphite has no particles estimated to be larger than 1/330,000 inch in diameter. It is, of course, this finely divided graphite which is most generally used for lubricating purposes, and from the results of which the value of graphite as a lubricant will be judged.

On examination under a high-powered microscope any metal, no matter how carefully ground, polished or burnished, will be found to show many small irregularities, which are the real cause of friction. It is the scraping of these, one over another, and the constant cutting or wearing, which is so productive of hot boxes, cut valves and cylinders. The thin, tough particles of graphite attach themselves to, and build up these irregularities, filling up the hollows and forming a thin but practically indestructible veneer-like coating of extreme smoothness. When formed,

under no circumstances short of the introduction of some foreign body which would cause scoring before the graphite began its equalizing work, could the metal surfaces come into actual contact with each other. The film being indestructible, no sudden hammer effect, such as takes place when the load is suddenly switched off a large generator and the armature shaft drops from its floating position, can do damage which might be done were oil used as a lubricant. Oil can easily be hammered out, but there is no known means whereby graphite can be knocked out of position in a bearing.

It is noteworthy that flake graphite may be introduced to cylinders, may pass into feed water, may even be purposely placed in the boilers without it doing the slightest injury. As a matter of fact, its presence is an advantage rather than a detriment, as the flakes becoming attached to the plates, prevent scale from taking hold, and so considerably lessen pitting and scaling. Also the heat transfer will not be affected as the substance is a particularly good conductor. For ordinary purposes graphite may be mixed with non-fluid oils, or even with soap-base greases with advantage, but obviously for valve and cylinder lubrication of condenser engines graphite of the purest possible quality should be used by itself or else the old trouble of separating the oil from the feed water would recur.

Graphite may be applied in the following manner: Whenever valves or cylinders are overhauled, graphite mixed with vaseline may be applied to the surfaces, as this will ensure a general distribution. Graphite can be introduced on starting up, through an oil cup or indicator pipe. It can be put in in a dry state, but a better plan is to float it in water and apply it in exactly the same way as lubricating oil is applied. If at any time there should be slight squeaking, or a bearing should start to warm up, a little graphite and water will effect an instant cure.—"Cassier's Engineering Monthly."

SUCCESSFUL MANUFACTURE OF RADIUM.

The cost of one gram of radium metal produced in the form of bromide during March, April and May of the present year was $36,050, I am informed by Dr. Charles L. Parsons, in charge of the radium investigations of the Bureau. This includes the cost of ore, insurance, repairs, amortization allowance for plant and equipment, cost of Bureau of Mines coöperation, and all expenses incident to the production of high-grade radium bromide. When you consider that radium has been selling for $120,000 and $160,000 a gram, you will see just what the Bureau of Mines has accomplished along these lines.

The cost of producing radium in the small experimental plant during the first few months of the Bureau's activities was somewhat higher, but not enough to seriously affect the final average.

The public, however, should not infer that this low cost of production necessarily means an immediate drop in the selling price of radium. The National Radium Institute was fortunate in securing, through the Crucible Steel Co., the right to mine 10 claims of carnotite ores belonging to them, and this was practically the only ore available at the time.

Since then new deposits have been opened, but these are closely held, and according to the best judgment of the experts employed by the Bureau of Mines the Colorado and Utah fields, which are much richer in radium-bearing ores than any others known, will supply ore for a few years only at the rate of production that obtained when the European war closed down the mines. The demand for radium will also increase rapidly, for the two or three surgeons who have a sufficient amount of this element to entitle them to speak from experience are obtaining results in the cure of cancer that are increasingly encouraging as their knowledge of its ap-

plication improves. A few more reports like that presented to the American Medical Association at its recent San Francisco meeting and the medical profession as a whole will be convinced of its efficacy. Under all the circumstances that have come to my knowledge it does seem to me that it behooves the Government to make some arrangement whereby these deposits, so unique in their extent and their richness, may be conserved in the truest sense for our people, by extracting the radium from the ores where it now lies useless and putting it to work for the eradication of cancer in the hospitals of the Army and Navy and the Public Health Service.

The 10 carnotite claims being operated at Long Park, Colo., by the National Radium Institute have already produced over 796 tons of ore averaging above 2 per cent. uranium oxide. The cost of ore delivered at the radium plant in Denver has averaged $81.30 per ton. This included 15 per cent. royalty, salary of Bureau of Mines employees, amortization of camp and equipment, and all expenses incident to the mining, transportation, grinding and sampling of the ore.

A concentrating plant for low-grade ores has been erected at the mines and is successfully recovering material formerly wasted. Grinding and sampling machinery has been installed at Denver and a radium-extraction plant erected in the same city. The radium plant has now a capacity of three tons of ore per day, having been more than doubled in size since last February. Before that time the plant has been run more or less on an experimental scale, although regularly producing radium since June, 1914. To July 1, slightly over three grams of radium metal has been obtained in the form of radium barium sulfate, containing over one milligram of radium to the kilogram of sulfates. The conversion of the sulfates into chlorides and the purification of the radium therefrom is easily accomplished and with very small loss of material. Unfortunately, however, special acid-proof enamel ware, obtainable only in France, has not been delivered of sufficient capacity to handle the crystallization of the full plant production, so that a little less than half the output, or, to be exact, 1,304 milligrams of radium element have been delivered to the two hospitals connected with the National Radium Institute. The radium remaining can be crystallized at any time from neutral solution in apparatus already installed, but the greater rapidity and efficiency of production of this very valuable material by the methods used have decided the Bureau of Mines to await the completion of apparatus now being built before pushing the chloride crystallization to full capacity.

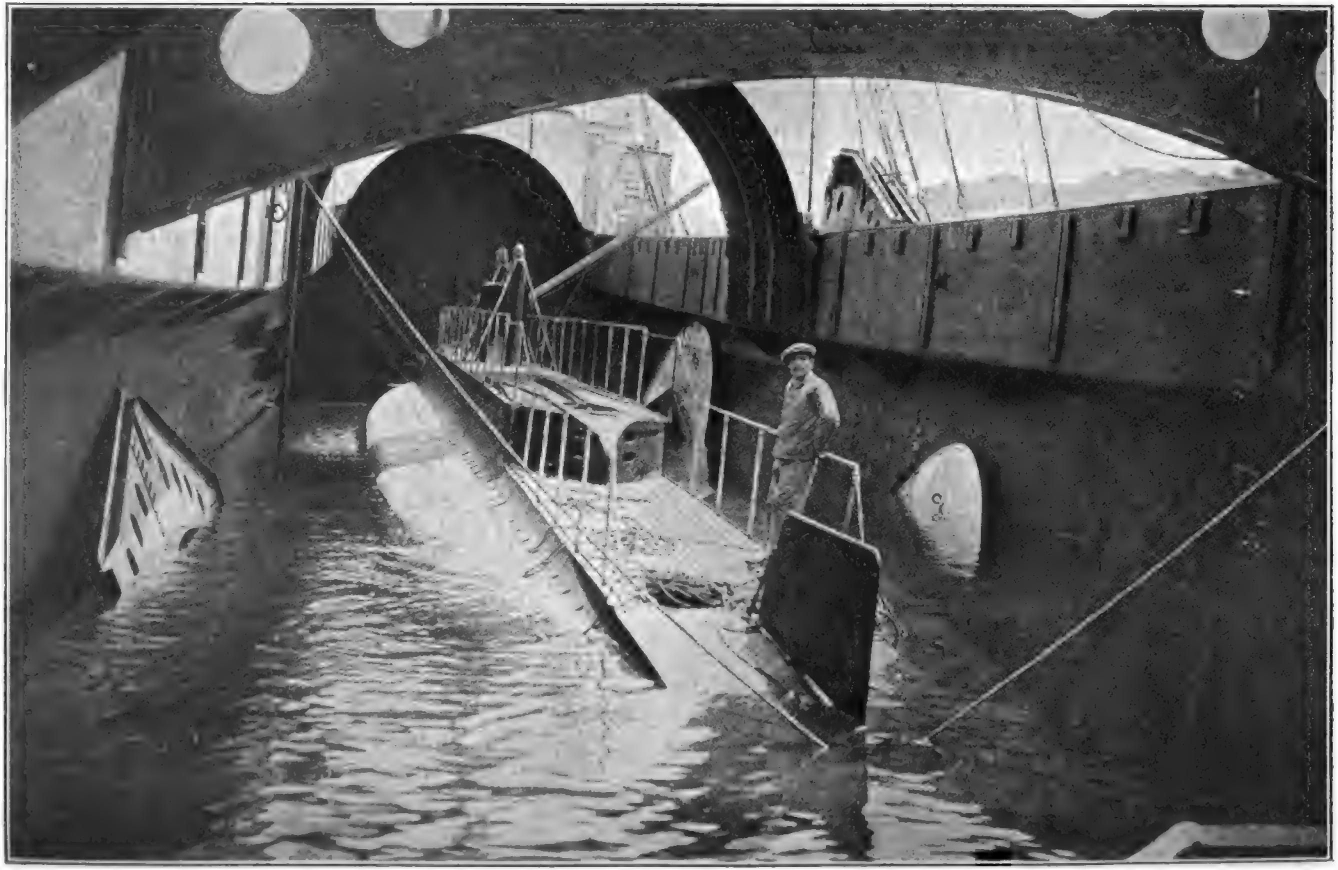
The average radium extraction of all ore mined by the National Radium Institute has been over 85 per cent. of the amount present in the ore as determined by actual measurement. The amount present in the ore has been found, in fact, to be essentially the same as the theoretical amount required by the uranium-radium ratio. The extraction figures for the last five carloads of carnotite treated has shown a recovery of over 90 per cent. in each case.

A bulletin giving details of mining, concentration and methods of extraction is being prepared by the Bureau of Mines and will be issued early in the fall.—Extract from U. S. Commerce Report No. 175, July 28, 1915.

ADAPTING A SUBMARINE TRANSPORTER FOR GENERAL CARGO.

The announcement that Messrs. Norton, Lilly & Co. have chartered the *Kanguroo* to load at New York for Bordeaux is an interesting sidelight on the demand for tonnage at present, since the *Kanguroo* was designed by Messrs. Schneider & Co., Le Creusot, for delivering submarines to foreign owners. The vessel, however, has the advantage that when not being used for this purpose it may be considered as a single-deck cargo boat,

THE "KANGUROO" WITH BOW PLATES REMOVED AND FRAMING ON PORT BOW CLEARED



SUBMARINE BERTHED AND AWAITING THE RAISING OF THE VESSEL TO CLEAR WATER OUT OF HOLD.

Submarine Preparing to Enter the "Kanguroo."

Another View of the Bow, and the Submarine Entering the "Kanguroo."

although it is obvious that the compromise effected in the design would hardly make the vessel a paying one, excepting in the existing high freight rates.

The *Kanguroo* has been employed as a submarine carrier on few occasions. The first two voyages were when Messrs. Schneider & Co. sent Laubeuf submarines to Callao for the Peruvian government, and last year she was employed by the Fiat San Giorgio, of Spezia, to deliver a Laurenti submarine to Rio de Janeiro.

The principal dimensions are: Length, extreme, 305 feet; breadth, 39 feet; draught (mean), 19 feet; displacement, 5,540 tons; carrying capacity, 3,830 tons; tonnage, 2,493 gross, 1,720 net.

The vessel is built entirely of steel and is propelled by triple-expansion engines of 850 H.P., giving a speed of about 10 knots. The ship is divided into three main parts, namely: (i) the after portion, containing the propelling and auxiliary machinery, accommodation for crew and so on; (ii) a large hold, 190 feet long, and capable of accommodating a submarine; and (iii) the fore part of the ship, specially designed for allowing the submarine to pass into the hold and containing the main tanks for raising and lowering the vessel.

This large hold consists of a double shell, with a hatchway covered by hatches removable by means of a traveler bridge. The space between the two skins is constructed for water ballast in the lower part and has air chambers in the upper portion.

In the bow of the ship there is a tunnel, the lateral walls of which are watertight. The space between these walls and the shell consists of three water-ballast tanks, one at the bottom and two higher up the sides. At the extremity of the tunnel, which is separated from the hold, strictly so-called, is a watertight double-swing door. The portion immediately in front of the entrance of the tunnel consists of sections which can be removed in order to admit the submarine.

The *Kanguroo* with bunker coal, has her trim so altered by the filling of the water-ballast tanks aft that the stem rises completely out of the water. The bow plates which cover the tunnel referred to are removed, and the double-swing doors at the extremity are opened, thus giving free access to the hold. The water tanks in the fore part of the ship are next filled. This has the effect of making her sink at the bows until the bottom of the hold is lower than the draught of the submarine. When this is completed and the vessel is trimmed satisfactorily the submarine enters the hold through the tunnel. Wooden chocks and stays are placed in the hold for the submarine to rest on, and when all is completed the trim of the *Kanguroo* is again altered by emptying the ballast-tanks in the fore part of the ship, whilst still maintaining communication between the hold and the sea. As soon as the stem of the ship is sufficiently out of the water the swing door is closed, the bow plates are replaced, leaving the submarine high and dry, resting on the chocks referred to. The trim of the vessel is then adjusted for proceeding to sea. These operations are here illustrated, but it will be seen that for certain types of cargo the large free hold is particularly advantageous. Moreover, there are cases, such as in the timber trade, where the facilities afforded by the open bow may be taken advantage of.—"Shipbuilding and Shipping Record."

AUXILIARY NAVAL VESSELS.

CLASSIFICATION OF NAVAL VESSELS—DESCRIPTION OF THE DESTROYER TENDER "MELVILLE."

The great variation in the composition of groups and types of vessels in the naval establishment of a first-class power may, perhaps, be conceived by scrutinizing the list appended below, comprising, it is believed, all the units as they would appear in active service:

A.—Fighting Ships.—(1) Battleships of the first line. (2) Battleships of the second line. (3) Armored cruisers. (4) Cruisers of the first class. (5) Cruisers of the second class. (6) Cruisers of the third class. (7) Destroyers. (8) Torpedo boats. (9) Submarines. (10) Gunboats.

B.—Coast and Harbor Defense Ships.—(1) Monitors. (2) Coast defense ships.

C.—Dispatch Ships.—(1) Scouts. (2) Converted yachts.

D.—Auxiliary Ships.—(1) Supply ships. (2) Hospital ships. (3) Oil fuel ships. (4) Colliers. (5) Transports. (6) Mine layers. (7) Mine sweepers.

E.—Miscellaneous Ships.—(1) Tugs. (2) Coal barges. (3) Ash lighters. (4) Water barges. (5) Ammunition lighters. (6) Freight lighters. (7) Floating workshops. (8) Floating derricks. (9) Garbage lighters. (10) Ferry boats. (11) Mud scows. (12) Dredges. (13) Diver barges.

The vastness of the scope for administration, operation, design and construction covering such a complexity of units, will be conceded when a little thought is bestowed on the fact that each type of the vessels enumerated represents an individual design made to fulfill a given purpose and to perform a certain predetermined work during operations.

Under the name "Supply Ships" as given in the list, there are to be found two separate more or less distinctive classes, viz.: A. Tenders for submarines. B. Tenders for destroyers.

Of the former type of recent construction the submarine tender *Fulton* was described in the issue of July, 1914, of this journal, and as representative of the latter class the present article will deal with the destroyer tender *Melville.*

It is a well-known fact that the dimensions of destroyers or submarines are as yet not of such proportions as to render them independent in their actions for any length of time, when cut off from access to supply depots. It might properly be mentioned that as a result of the limited capacity of such vessels, together with the very exacting demands placed upon them in service, the construction of tenders or supply ships was essentially born.

SERVICE CHARACTERISTICS.

As the service intended to be performed by ships coming under this class of vessels is to supply munitions of war, provisions and stores, the interior arrangements are made primarily with a view to meet this purpose. There are also arrangements made on board to care for a large number of sick or otherwise incapacitated.

Among the supplies which a tender will be called upon to provide we find the following: A. Air, compressed and carried in flasks. B. Ice and refrigerated supplies. C. Light and electric current. D. Fresh water. E. The output from machine shops for necessary repairs. F. Hospital service (for which provision of 38 beds has been made).

Besides machinery and output for the supply of foregoing necessities, storage is provided for torpedoes, mines and ammunition, meats, stores and provisions.

GENERAL FEATURES.

The general appearance of the new destroyer tender *Melville,* with its two masts and straight stem, somewhat resembles an ocean liner, but becomes distinctly military in the light of the batteries, consisting of eight 5-inch 51-caliber rifles and two 3-pounder saluting guns.

The general dimensions and data pertaining to the ship are as follows:

Length between perpendiculars, 400 feet; breadth, 54 feet 5½ inches; depth, 20 feet; displacement (normal), 7,150 tons; oil fuel carried, 900 tons; speed, in knots, 15; total horsepower, main turbines, 4,000; number of shafts and screws, one; type and number of boilers, two B. & W.; total heating surface, square feet, 7,000; three generating sets, 100 kilo-

watts each, 125 volts; total complement of officers and crew, 254; name of builders, New York Shipbuilding Company; launched, March 4, 1915; state of finish, 90 per cent.; cost of vessel, complete, $1,310,000 (£268,000).

GENERAL HULL ARRANGEMENT.

There are three principal decks, consisting of main deck, second and third deck. Below these decks forward is placed a platform deck which extends aft as far as possible and also partially amidships. Below the platform deck is the hold, divided into suitable compartments, among which are trimming tanks, hold space, stores and space for any provisions. On the deck spaces above are to be found special storerooms, cold storage rooms and magazines, officers' quarters aft and crew space forward. On the main deck are located three deck houses. In the forward deckhouse are arranged, besides the commanding officer's cabin and office, executive offices and navigating officer's stateroom. Similarly in the after deckhouse there are located the junior officers' staterooms and the flotilla commander's office and cabin. Within the deckhouse placed amidships there are situated the engine-room hatchway, coppersmith and blacksmith shops, a foundry, the general storekeeper's and paymaster's office and a trunk for the smoke pipe, together with various other smaller compartments.

MACHINERY ARRANGEMENT.

The driving power in this ship is developed by steam turbines, while in the submarine tender *Fulton* the corresponding power is produced by a Diesel engine.

The engine room is divided below the main deck into three compartments of which the center compartment constitutes the turbine room, while each wing compartment constitutes an auxiliary engine room. There are located in the starboard engine room three ice machines, the auxiliary condenser, and above on a gallery, the generators with switchboard. In the port engine room below the gallery are to be found the main condenser, the main air and circulating pumps and feed tanks, and on the gallery four double-effect evaporators, together with the necessary pumps.

Forward of the engine compartment is the boiler compartment, where are placed, besides the boilers, fuel-oil settling tanks, pumps and blowers.

The capacity of the evaporating plant is about 25,000 gallons for each twenty-four hours, and each of the three ice machines will yield 3 tons of ice every twenty-four hours. The three turbo generators are each of 100 kilowatt capacity.

There are placed in the machinery spaces numerous other auxiliaries, such as the main and auxiliary feed pumps, fire and bilge pumps, lubricating-oil pumps, evaporator feed and fresh-water pumps, fuel-oil supply pumps and burner pumps. The distillers are placed at a higher level in the engine-room hatch.

The turbine machinery is of Parsons type and consists of two separate turbine units, of which the high-pressure unit is placed on the starboard side and the low-pressure unit on the port side of the ship.

The transmission of power from turbines to the propeller is effected by means of a Westinghouse mechanical reduction gear, consisting substantially of a pinion on each turbine shaft meshing with a gear wheel attached to the center shafting driving the propeller. The propeller revolves at a speed of 110 revolutions per minute, while the turbines run at 1,400 revolutions. The speed of transmission is thus reduced in a ratio of 140 to 11, or nearly thirteen times.

The gearing is of the double-spiral type with the teeth of each cut in the opposite direction. Each pinion transmits 2,000 horsepower, and, being of the same diameter, revolves at the same speed, viz., 1,400 revolutions per minute. The gear-wheel center is made of cast iron, the rim of

cast steel and the shaft of mild-steel forging, while the pinions are made of rather high-grade steel forgings and the pinion-driving shafts of chrome-nickel steel. The angle of the spiral is about 30 degrees and the width of face is 16 inches for each single gear, making a total width of driving face of 32 inches with 40 tooth contact points with a contact pressure of 562 pounds.

The general advantages claimed for transmission of the propulsive power by means of reduction gearing as compared with straight turbine drive are:

1. About 15 per cent. increase in the economy of coal consumption.
2. Average reduction in the weight of the machinery about 15 per cent.
3. The reduction in the size of turbine rotors, which is of practical advantage in handling, as well as general reduction in the size of turbines.

To meet the requirements for successful operation, the gearing must be durable, practically noiseless, with the transition of pressure from tooth to tooth without shock.

The bearing supports of the pinions for mechanical gearing are made sometimes rigid and sometimes with floating arrangements, both of which have given satisfaction. The Westinghouse gearing is provided with a hydraulic floating arrangement for each pinion, and is now made for several United States vessels.—"International Marine Engineering."

HALL'S SYSTEM OF REFRIGERATION FOR SHIPS.

The accompanying illustrations show the latest design of refrigerating machinery for ships, which is manufactured by Messrs. J. & E. Hall, Ltd., Dartford Iron Works, Kent.

This firm was founded by John Hall in 1785, that is, 130 years ago, and they have been pioneers in mechanical progress. Their machines are on CO_2 (carbonic anhydride) system, for which they claim the following special advantages: Although dense carbonic anhydride (CO_2) will not support life, it is, in the quantities used in a refrigerating machine, quite innocuous. The whole charge may be (and has been experimentally) allowed to escape in an engine room without ill effects or inconvenience being experienced by the attendants. Ammonia (NH_3) and sulphurous acid (SO_2), on the other hand, are actively poisonous gases; a very small percentage in the air is inimical to life, so that any considerable leakage is likely to have serious consequences, and many fatal accidents have been attributed to this cause. The harmless nature of CO_2 permits of a safety valve being fitted to relieve any excessive pressure which might be created through neglect or oversight. Carbonic anhydride (CO_2) is tasteless and inodorous, and has no effect, detrimental or otherwise, upon any goods, even the most delicate and easily-tainted foodstuffs, with which it may come in contact. It is, in fact, the harmless gas used for the "aeration" of mineral waters, wines and beer. CO_2 has no affinity for metals, consequently the materials that are most suitable can be adopted for the various parts of the machine. For example, copper is used for the condenser coils of marine machines because of the corrosive action of sea water upon iron. These copper coils are indestructible and retain their efficiency and their intrinsic value after many years of service. Whilst the carbonic anhydride (CO_2) machine compares favorably in economical working with any other system, it has the advantage of employing a cheap and easily-obtainable refrigerating medium. For example, the cost of a complete charge of CO_2 for a machine of 24 tons ice-making capacity is about £5, compared to £112 for a charge of ammonia.

Referring to the diagram, Fig. 2, the principle will be readily under-

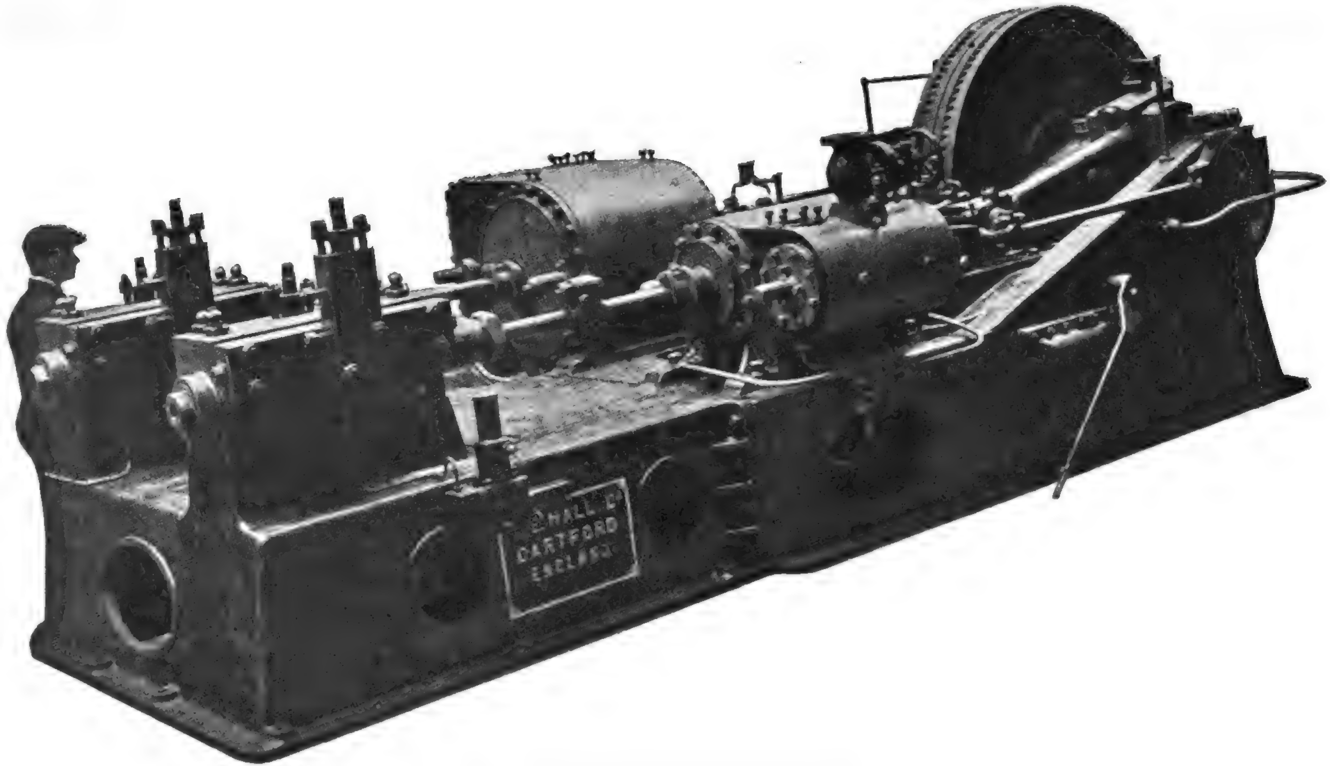


FIG. 1.—HORIZONTAL DUPLEX MACHINE.

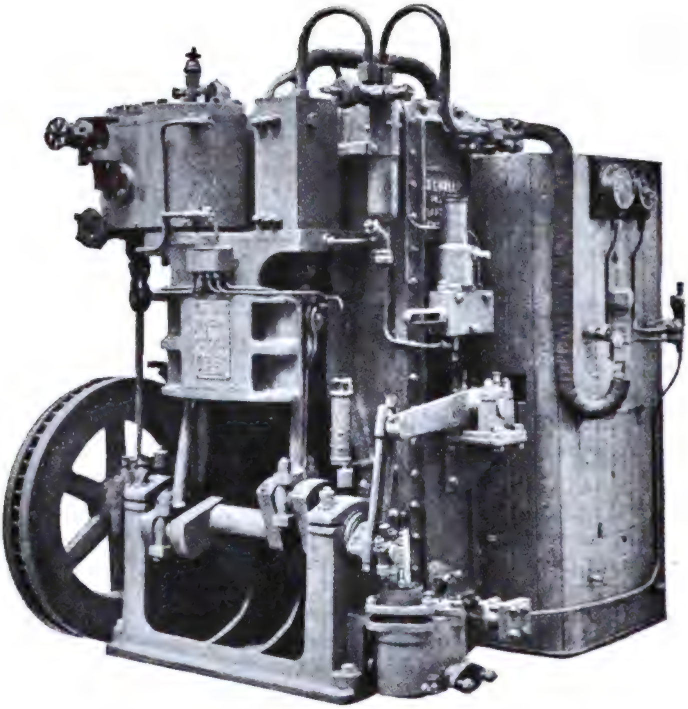


FIG. 3.—VERTICAL SINGLE MARINE-TYPE MACHINE.

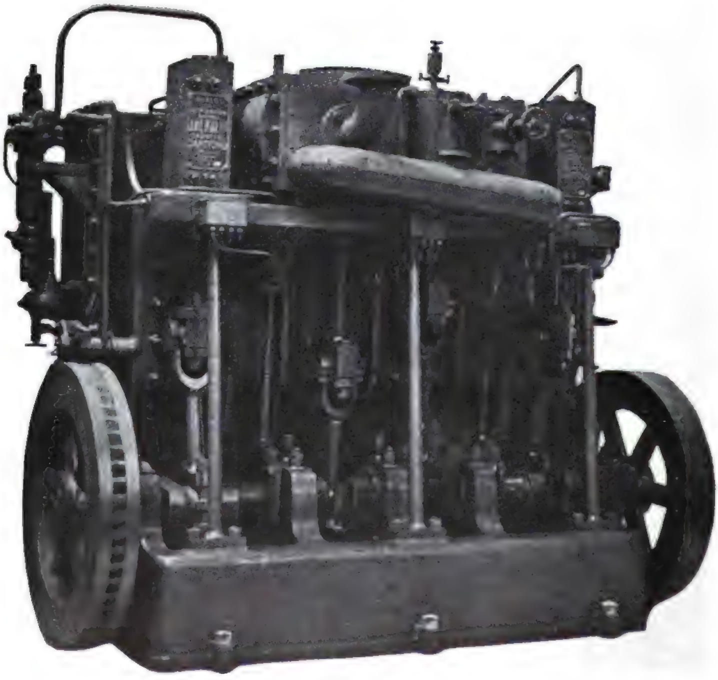


FIG. 4.—VERTICAL DUPLEX MACHINE.

stood. The CO_2 machine is one of that type of refrigerating machines wherein the refrigerant during the cycle of operations undergoes a change of state, from gas to liquid and from liquid to gas, the transformation being accompanied by the liberation and absorption of a quantity of heat many times greater than that due to the mere change of temperature of the refrigerant. This type of refrigerating machine consists essentially of the following three parts, shown diagrammatically in the illustration, Fig. 2: The compressor, in which gas drawn from the evaporator is compressed; the condenser, consisting of coils in which the hot gas delivered from the compressor is cooled and liquefied by the action of the cooling water; the evaporator, consisting of coils in which the liquefied CO_2 is evaporated by heat abstracted from the material to be cooled, producing any degree of temperature that may be required down to 80 degrees F. below freezing point. The simplest conception of the refrigerating machine is that it is a heat pump, *i. e.*, heat is abstracted at the "low level" of the cooling duty performed and discharged at the "high level" of the condensing water. The cycle of operations is as follows: On the suction stroke of the compressor piston a charge of gas is drawn from the evaporator and on the return stroke the gas is compressed and discharged

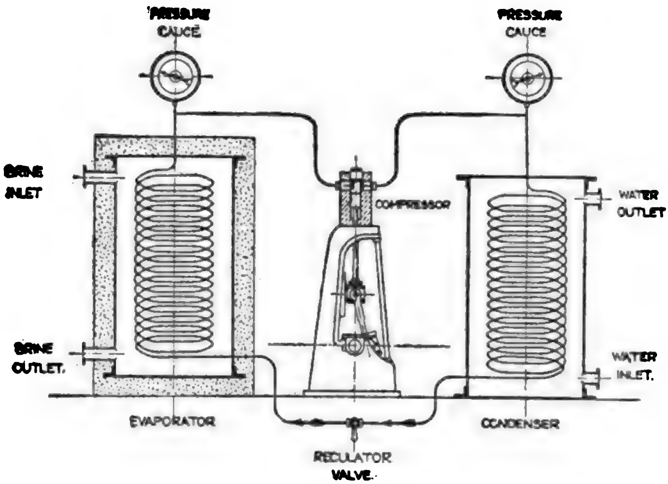


FIG. 2.—DIAGRAM SHOWING WORKING.

into the condenser coil at a pressure sufficient to cause liquefaction, this pressure depending directly on the temperature of the cooling water. The gas enters the condenser coils at a high temperature and during its passage through the coils is cooled to a temperature within a few degrees of the cooling water circulating over the surface of the coils, with the result that the gas is liquefied, the latent heat of liquefaction passing into the surrounding water. The liquid CO_2 then passes from the lower terminals of the condenser coils through the regulating valve to the evaporator coils, where evaporation takes place at a temperature sufficiently below that of the medium to be cooled to allow of a reasonably rapid interchange of heat. The evaporator coils thus perform the reverse function of the condenser coils, the heat required to evaporate the liquid CO_2 being absorbed from the surrounding brine solution or other medium, which is consequently lowered in temperature. The outlet terminals of

the evaporator coils are connected to the suction of the compressor, the CO_2 passing in the form of a gas from the evaporator to the compressor, and thus completing the cycle. The action of the evaporator of a refrigerating machine is analogous to that of a steam boiler, the liquid CO_2 taking the place of water as the medium to be evaporated and the surrounding solution of brine replacing the fire as the source from which the heat required to effect evaporation is supplied. Just as water under atmospheric pressure boils at a temperature of 212 degrees F., and under a pressure of 100 pounds per square inch at a temperature of 337 degrees, so liquid CO_2 under a pressure of 21 atmospheres boils at a temperature of zero F., and so on for other pressures. Under atmospheric pressure CO_2 evaporates from the solid at the extremely low temperature of 120 degrees below zero F., or 152 degrees below the freezing point of water. The production of so low a temperature does not, of course, come within the scope of the commercial refrigerating machine, but this property has frequently been made use of in the laboratory for experimental purposes in connection with the liquefaction of the more permanent gases. The charge of carbonic anhydride originally put into the machine is used over and over again, going progressively through the processes of compression, condensation and evaporation. Thus a small quantity only is required to be added from time to time to replace any small losses. The following general description of marine CO_2 machines will make their construction clear: The compressors are bored out of solid forgings of high-carbon steel on account of the freedom of that material from porosity and because it provides a highly polished working surface for the cup leathers with which the pistons are usually fitted. The compressor gland is made gas-tight by means of cupped leathers embracing the compressor rod, into the space between which is forced a special oil by means of an automatic-pressure lubricator, the result being that a pressure superior to the greatest pressure in the compressor is maintained. Any little leakage past the leathers is a leakage of oil, either into the compressor or out into the atmosphere, and not a leakage of gas. What little leakage of oil takes place into the compressor is advantageous inasmuch as it both lubricates the compressor and also fills up all clearances, thereby increasing the efficiency of the compressor. The pressure lubricator is fitted with a tell-tale to indicate the quantity of oil remaining, and also with a small hand pump, a few strokes of which are occasionally required to recharge the lubricator. In certain cases, and when specially asked for, a simple but very effective metallic-packed gland can be supplied. Between compressor and condenser the gas passes through a separator by which any excess of oil is impounded and should be drawn off at intervals. Such oil can be re-used after filtration. The condenser consists of nests of solid-drawn copper coils in which the gaseous CO_2 is liquefied. They are strapped rigidly together by means of stays and bolts or rivets, all of bronze. Each length of pipe is specially brazed to the next to form a continuous coil without joints. The condenser coils, round which sea water is circulated, are usually contained in the base of the machine or in an independent cast-iron tank strongly ribbed. The evaporator consists of nests of wrought-iron hydraulic pipes electrically welded into long lengths, and inside them the liquid carbonic anhydride evaporates. The heat required for evaporation is usually obtained either from brine surrounding the pipes, as in cases where brine is used as a cooling medium, or else from air or water surrounding the pipes, as in cases where it is required to cool air or water direct. A regulating valve is placed between the condenser and evaporator for adjusting the quantity of the liquid carbonic anhydride passing from the condenser. In order to enable the compressor to be opened up for examination of valves and piston without loss of carbonic anhydride it is necessary to fit a stop valve on the suction and delivery sides, so as to confine the carbonic anhydride to the condenser and evaporator. It is,

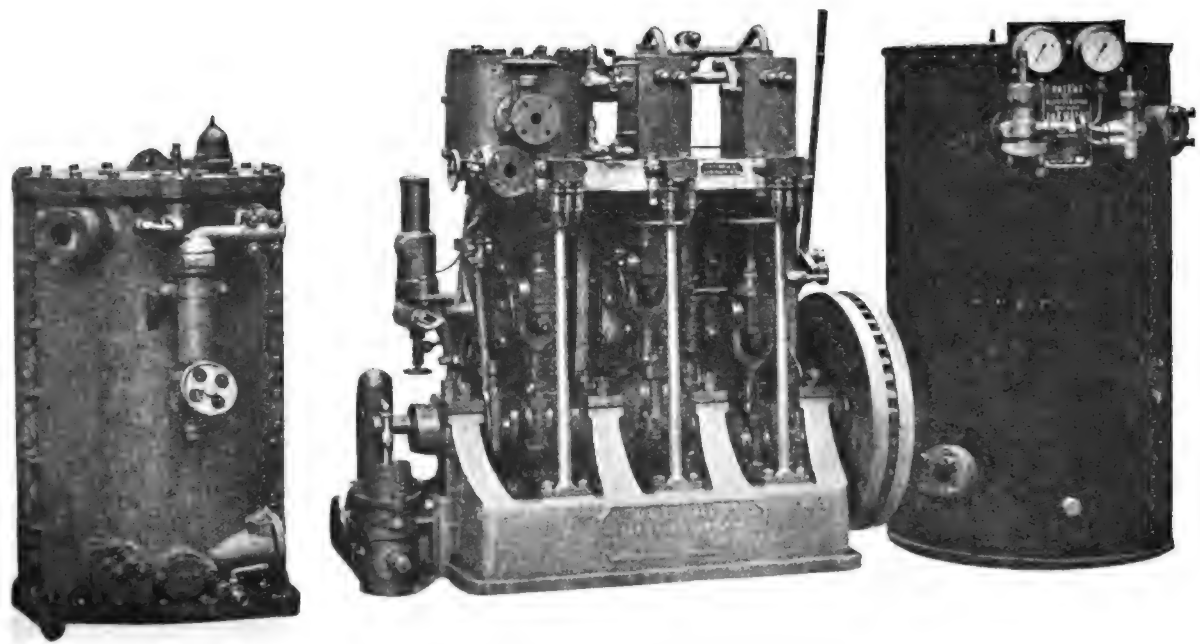


FIG. 5.—NAVAL-TYPE STEAM-DRIVEN MACHINE.

of course, possible for a careless attendant to start the machine again without opening the delivery valve. In such case an excessive pressure would be created in the delivery pipe, from which there would be no outlet. To provide against this danger a safety device (see illustration) is adopted, consisting of an ordinary spring safety valve, at the base of which is a thin copper disc designed to relieve at a certain pressure, considerably below that to which the machines are tested. This disc is made perfectly gas-tight, which could not be effected by the spring safety valve alone, the latter only coming into play when the disc is ruptured. Great care is necessarily exercised in making the discs to provide against variation in strength, due to any want of accuracy either in the thickness or hardness of the copper sheets out of which the discs are made. Tightness of joints is obtained with entire success by the use of rings turned from solid-drawn copper tube which withstand the heat and yet have the necessary elasticity to ensure the joint being perfectly tight whether hot or cold. Nevertheless, as an additional precaution, every CO_2 joint throughout the machine is always made accessible. During manufacture, all parts of the machine subject to the pressure of the refrigerant are tested for strength by hydraulic pressure to three times the highest working pressure, and they are again tested for detecting any possible porosity by air pressure at 90 atmospheres while submerged in water. Owing to the comparatively small diameter of all parts, even in large machines, there is no difficulty in securing a very ample margin of strength.

The illustration, Fig. 3, shows the vertical single marine-type machine. This type of machine is made in three sizes, 7, 8 and 8A. The compressor, made from a solid forging of high-carbon steel, is driven through a two-throw crankshaft by a single-cylinder steam engine, the compressor and steam cylinder being mounted side by side on a strongly ribbed cast-iron frame. The back of the frame is arranged as a casing to contain the CO_2 condenser coils, which are of solid-drawn copper tube with brass stays and straps. The water-circulating pump, made entirely of gunmetal, is attached to the main frame and worked off a pin on the end of the crankshaft. A cast-iron casing containing the wrought-iron CO_2 evaporator coils is bolted to the back of the CO_2 condenser. This casing is insulated with granulated cork retained by tongued and grooved boards neatly finished. With the No. 8A machine an independent duplex steam-driven brine pump is provided for brine circulation. The No. 7 and No. 8 machines have the brine pump attached to the evaporator casing and insulated with it and driven off the crankshaft by means of a sway beam. Independent brine pumps can be supplied if required. A very large number of these machines have been supplied for preserving passengers' provisions, and almost every important line of passenger steamers have ships fitted with them.

The illustration, Fig. 4, shows a vertical duplex machine. This machine has its compressors carried on a strong cast-iron frame with forged-steel pillars. The steam engine is compound, and drives the compressors through a four-throw crankshaft, the crankshaft being carried in five main bearings. Crosshead guides are of the open type and very accessible. The two sets of condenser coils are of solid-drawn copper tube with brass straps and bolts, and are contained in separate compartments in the back part of the frame. Sea-water and brine-circulating pumps are of the independent steam-driven type. The two sets of electrically-welded wrought-iron evaporator coils are contained in a wrought-iron double casing, which can be specially made to suit the position in which it has to be placed.

The illustration, Fig. 1, shows a horizontal duplex machine. This is the type of machine fitted on large meat-carrying steamers and on some large fruit steamers. The steam surface condenser is arranged in the bed of the machine, with air and feed pumps driven off the crankshaft. The two sets of CO_2 condenser coils are each contained in a separate

rectangular cast-iron casing. The evaporator coils are contained in separate circular steel casings. The water-circulating pump is of the vertical duplex type.

The illustrations, Fig. 5, show the steam-driven naval-type machine. The frame is of cast iron, light but strongly ribbed, reinforced by polished steel columns in front. The compressors are made from solid forgings of high-carbon steel, and are of either the single-acting or double-acting type. The compressor rods are of nickel-steel, and the pistons are fitted with leather packing or metallic rings as required, similar packing being employed in the glands, which are fed with oil at a pressure above that below the piston by means of the differential lubricator. The crossheads are fitted with flat slipper guides, and the crankshaft of polished steel is carried in two, three or four main bearings according to the number of cranks. The sea-water circulating pump is of gunmetal and is driven by the crankshaft. The CO_2 condenser is of solid-drawn copper coils in a cast-iron casing, divided to facilitate overhaul in place. The evaporator coils, of steel or wrought iron, are contained in a similarly constructed cast-iron casing, which is well insulated and lagged with wood or galvanized steel. The brine pumps are independent, of the vertical duplex type with steam-driven machines and of the centrifugal type coupled direct to an electric motor with electrically-driven machines. Each plant is supplied complete with all necessary and usual accessories, also gas and brine-connecting piping between the various parts of the plant, brine-mixing tank, and a complete set of spanners and special tools.

—"Steamship."

THE SOUTHWARK-HARRIS DIESEL ENGINE.

SYNOPSIS.—A two-stroke-cycle engine with no scavenging or starting valves in the head, employing a stepped piston for both starting and scavenging and possessing unusual means of fuel control.

The impetus given the heavy-oil-engine industry in this country by the expiration of the original Diesel patents in 1912 has been marked particularly by the number of steam-engine builders who have entered this field. Some, choosing to follow foreign practice, are building under license from foreign firms with modifications to suit local conditions; others have developed what may be termed distinctly American designs. Among the latter may be mentioned the Southwark-Harris Diesel engine, built by the Southwark Foundry & Machine Co., of Philadelphia, from the designs of Leonard B. Harris.

This is a two-stroke-cycle type intended primarily for marine service, but also adapted to stationary work, the engine being somewhat simpler for the latter service, as no reversing is required. Because of the ingenious, yet simple and flexible, control for maneuvering, the marine type will be described.

Unlike most two-stroke-cycle Diesel engines, there are no starting nor scavenging valves in the head, the only opening being for the fuel atomizers, of which there is one to each cylinder in both marine and stationary types. There are two atomizer-actuating levers in the marine type, one for ahead and the other for astern; in the stationary type there is, of course, only one atomizer-actuating lever. This arrangement makes possible a very simple cylinder-head casting.

The pistons, as shown in Fig. 3, are stepped, the lower part serving as a scavenging pump besides acting as a guide in the place of a crosshead. The cylinders are in pairs, and each scavenging piston serves the adjacent cylinder of that pair through the passage A, valve V_2, manifold M and port D. This will be understood when it is remembered that the cranks

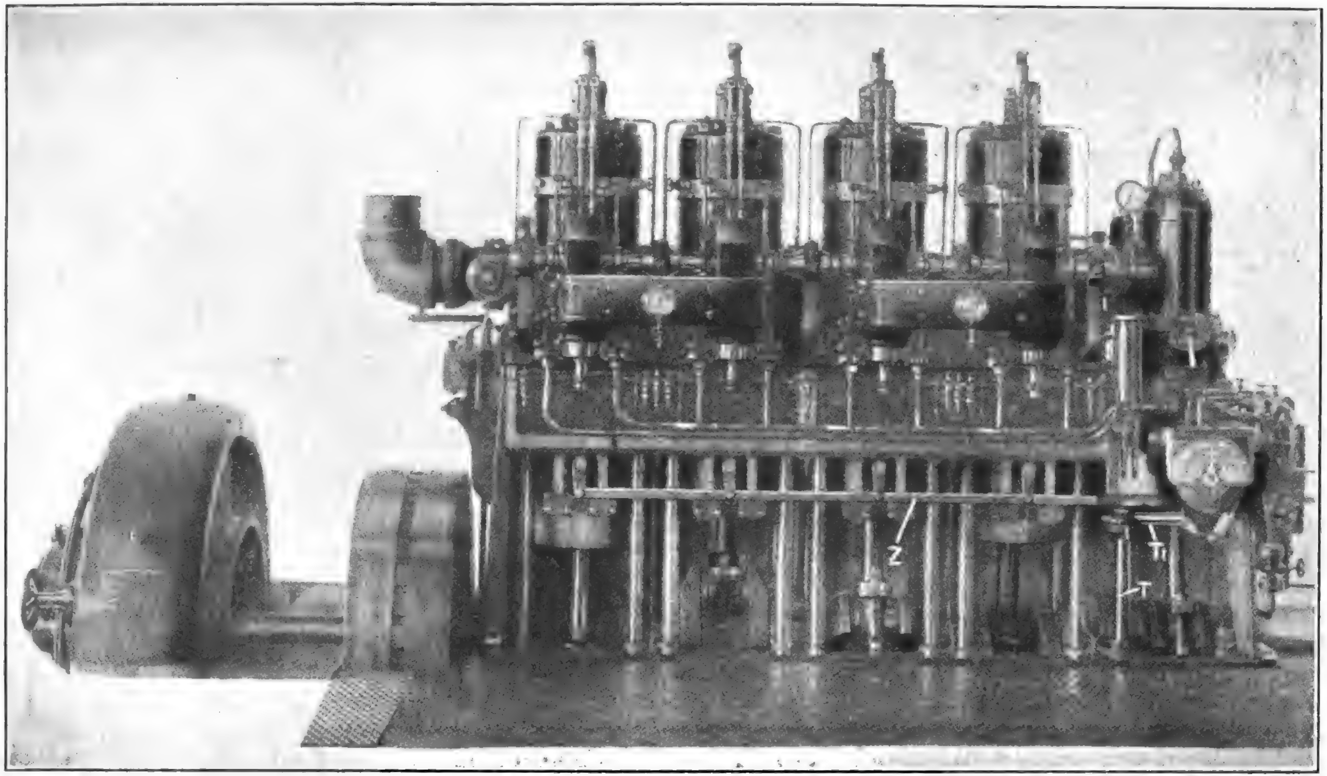


FIG. 1.—240-H.P. SOUTHWARK-HARRIS DIESEL ENGINE.

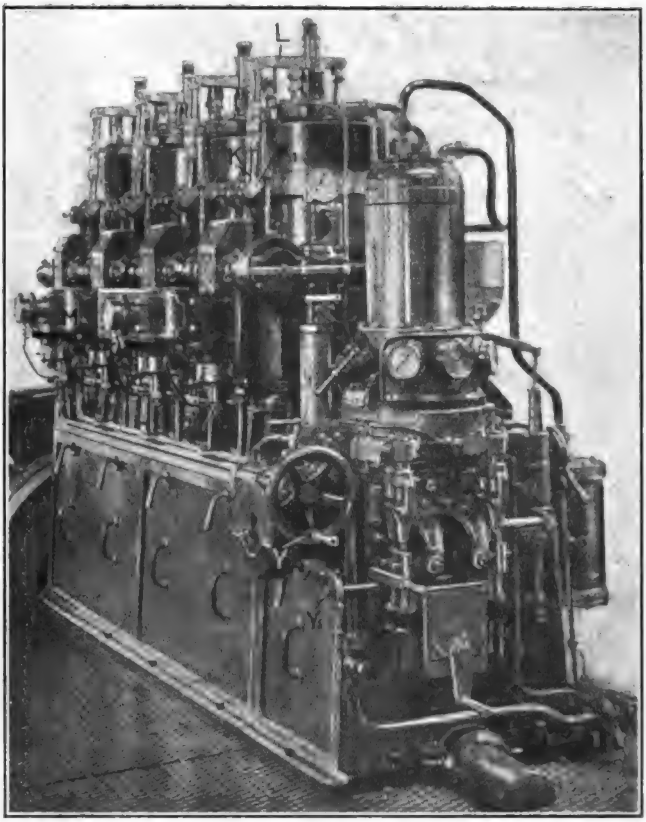


Fig. 2.—End View, Showing Pumps.

of a pair are set at 180 degrees. Therefore, when the scavenging piston of cylinder No. 1 is traveling upward, compressing the air in the scavenging cylinder, ports and manifold, the pistons of No. 2 are traveling downward, and when No. 2 working piston has uncovered its port D, the scavenging air, under a pressure of about 7 pounds, will rush in and force the spent products of combustion out through the exhaust ports E. The air-delivery valve of cylinder No. 2 prevents the scavenging air of No. 1 from being forced into the scavenging cylinder of No. 2 while its pistons are on the down stroke.

An unusual feature of this stepped piston is its use for starting the engine—a most important factor in marine work, as it avoids admitting cold starting air to the highly heated working cylinders and pistons when reversing and maneuvering. Moreover, as the area of the stepped piston is greater than that of the working piston, starting air of relatively low pressure, 175 pounds, can be employed. Since the starting is independent of the working cylinders, the fuel can be admitted to the latter while the starting air is still on. This will be found advantageous when starting under load, as the starting air can thus be used to help out until the momentum has been built up.

Of interest in this connection are the diagrams of Fig. 4. No. 1 is from the scavenging cylinder when starting with air, while No. 3 was taken simultaneously in the working cylinder. Line *a*, diagram No. 1, represents the first outward stroke, *b* the return stroke, and *c* the successive outward strokes until the starting air is shut off. It will be noted that after the first stroke, a pressure of only 30 pounds is required, owing to the power given back in the working cylinder by the expansion of the compressed air within it. Diagram No. 2 shows the normal working of the scavenging piston.

Starting and fuel-injection air is furnished by a two-stage compressor driven off the main shaft and having a control valve on the suction. The compressor delivers the high-pressure air for fuel injection directly to a steel air bottle mounted at the back of the engine frame. The starting air is supplied to the starting bottles through a reducing valve.

There is a separate fuel pump for each cylinder, making four in all in the engine shown. These are mounted at the end of the engine (see Fig. 2) and the stroke is varied by the governor.

A better idea of the operation of the fuel pumps will be gained by reference to Fig. 5. First, however, it will be necessary to revert to Fig. 1, which shows the pump shaft T_1 driven from the main crankshaft through the vertical shaft T and worm gears. At the end of T_1 is mounted a cam U, Fig. 5, which acts laterally upon the bell-crank levers W; these in turn work the pumps through the arms W_1, working in yokes on the pump stems. The fulcrum pins of the bell cranks are carried on laterally sliding plates V, the movement of which is effected through two worm splindles, each carrying a pair of right-and-left-hand worms meshing with sectors X. The upper worm spindle extends to the right and connects through gearing with the governor shaft, while the lower worm spindle extends to the left carrying the handwheel Y_1 and connecting through links and a rack (see Fig. 6) with the main control handle Y.

When the pointer of the main control handle is in the central, or "stop," position (see Fig. 1) the bell-crank levers W (Fig. 5) are separated so as not to be actuated by the cam U. Through the arrangement of links shown in Fig. 6, this position of the bell cranks is held ordinarily through the starting period. When the control handle is turned past the first notch to the running position the lower worm spindle is rotated and, by means of the sectors and slides, brings the bell-crank levers closer together, so that they are actuated by the cam U, and the fuel pumps are set in operation. The governor, now acting through the upper worm spindle, is able to control the position of the bell cranks and vary the

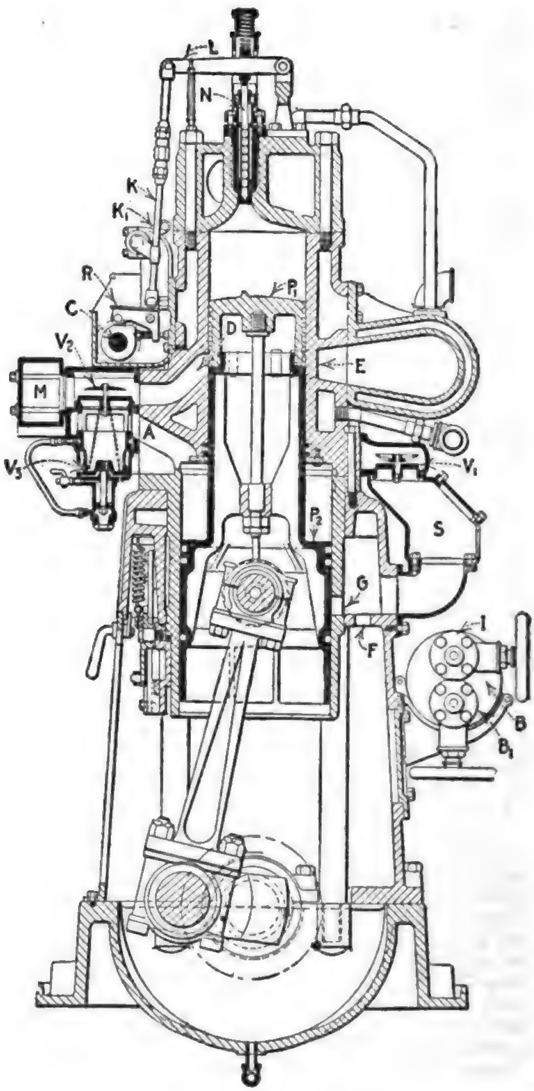


Fig. 3.—Sectional Elevation.

P_1 is the main piston; P_2, the scavenging and air-starting piston; V_1, scavenging-air inlet valve; A, scavenging air outlet passage and starting-air inlet passage; V_2, scavenging-air delivery valve; M, manifold; D, scavenging-air inlet ports; V_3, the air-operated intercepting valve; O, outlet for starting air; S, silencer; F, vents; B, injection-air storage bottle; C, cam-shaft; R, atomizer-actuating rockers; K, push rods; L, atomizer levers; N, atomizer spindle; E, exhaust ports; I, injection air from compressor; B, bypass to starting bottles; K_1, bell crank.

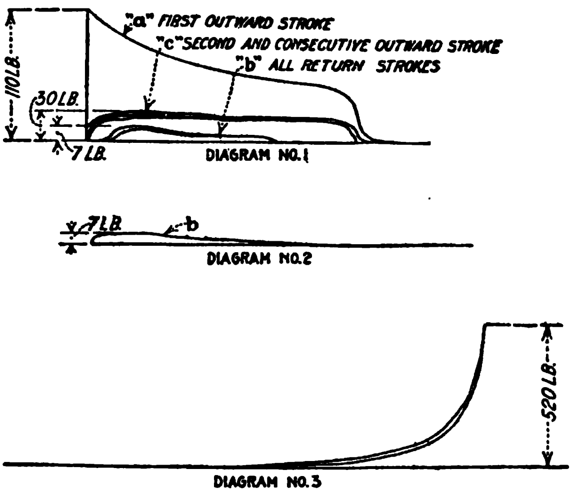


FIG. 4.—INDICATOR DIAGRAMS FROM WORKING AND STARTING CYLINDERS.

stroke of the pumps to suit the load. The small handwheel Y_1 permits manual control of the fuel without altering the position of the main control wheel Y. By this means the engineer is enabled to control the speed of the engine at will, and the governor will maintain control at this speed. This feature is especially useful in marine work when running through a heavy head sea with the engine racing.

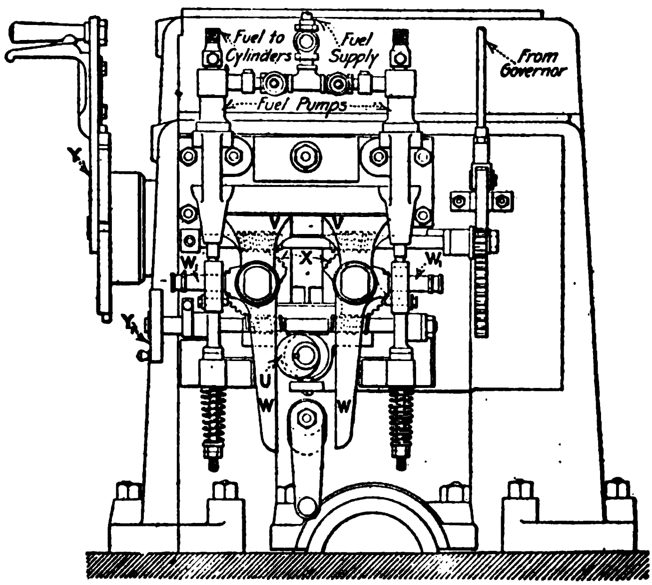


FIG. 5.—SHOWING FUEL CONTROL.

In addition to the fuel control through varying the stroke of the pumps, the lift of the fuel atomizers may be altered at will from the control wheel while the engine is running. Referring to Fig. 3, the lower ends of the atomizer push rods may be swung outward through the arc on the upper side of the rockers, which in turn are actuated by the cams on the main camshaft. It will be seen that when the push rods are at the extreme right the rockers can be actuated without imparting any motion to the atomizer spindles. When they are moved to the extreme left the atomizers will have their greatest travel. Intermediate positions of the push rods on the rockers will correspond with definite openings of the atomizers. These push rods are shifted by means of the bell cranks

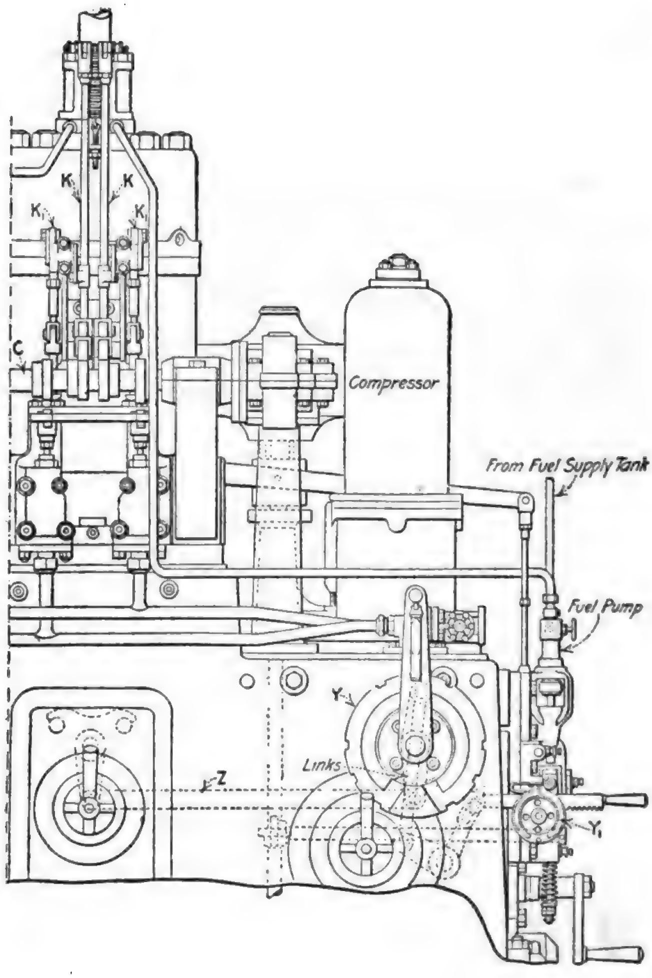


Fig. 6.—Partial Side Elevation, Showing Further the Control Features.

shown, which in turn are operated through vertical rods connected with the horizontal bar Z (Figs. 1 and 6), also connected with the control handle. The operation is obvious.

A four-cylinder, 240-I.H.P. Southwark-Harris engine has just been installed in the yacht *Southwark*, owned by C. P. Vauclain, of Philadelphia. The *Southwark* is 98 feet overall, 16 feet beam and 7 feet draught, and on her first trial trip made a speed of about 10 miles per hour against the tide and a head wind, with the engine turning up at 225 r.p.m. Extensive tests are now being made, and the results will be available at an early date.

The principal dimensions, horsepower, weight, etc., of the sizes listed are given in the following table:

PARTICULARS OF STATIONARY TYPE.

Indicated horsepower.	No. of cylinders.	I.H.P. per cylinder.	Cylinder diameter, ins.	Stroke, inches.	Revolutions per minute.	Approximate weight without flywheel, lbs.	Weight per I.H.P., lbs.	Floor space, square feet.	Length overall, feet and ins.
120	2	60	9	13	300	14,000	117	21.6	6–9
240	4	60	9	13	300	25,000	104	34	10–2
360	6	60	9	13	300	35,000	97	45	13–7
225	2	112.5	12	21	200	27,000	120	51	9–8
450	4	112.5	12	21	200	47,000	104	80	15–4
675	6	112.2	12	21	200	66,000	102	110	21–0
400	2	200	16	28	150	....	...	116	14–6
800	4	200	16	28	150	....	...	180	22–6
1,200	6	200	16	28	150	....	...	244	30–6
				MARINE TYPE.					
240	4	60	9	13	300	25,000	104	34	10–2
360	6	60	9	13	300	35,000	97	45	13–7
480	8	60	9	13	300	44,000	91.5	56.5	17–0
450	4	112.5	12	21	200	47,000	104	80	15–4
675	6	112.5	12	21	200	66,000	102	110	21–0
900	8	112.5	12	21	200	85,000	94.5	140	26–8
800	4	200	16	28	150	....	...	180	22–6
1,200	6	200	16	28	150	....	...	244	30–6
1,600	8	200	16	28	150	....	...	308	38–6

—"Power."

A MERCURY VAPOR ENGINE.

INCREASING THE EFFICIENCY OF A HEAT ENGINE BY EMPLOYING A LIQUID OF HIGH BOILING POINT.

To the layman the word *thermodynamics* suggests something abstruse, something perhaps remotely connected with practical things. Yet one of the fundamental conclusions from the principle of thermodynamics is as intensely practical as it is unexpected. The maximum efficiency of a heat engine working between a temperature t_2 of the boiler and t_1 of the condenser is wholly independent of the working substance (steam, hot

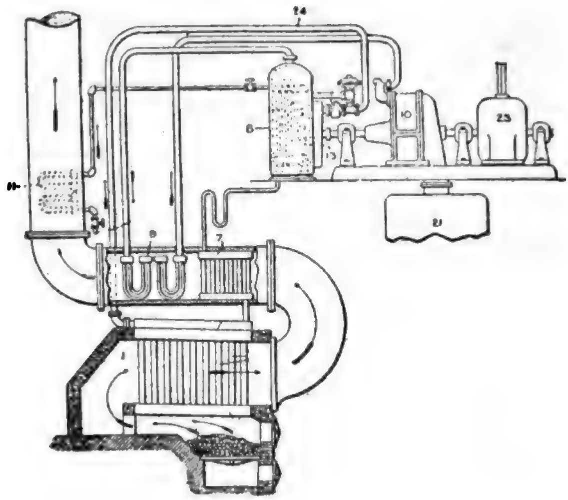


Fig. 1.—Diagrammatic View of Apparatus to Generate Power.

1, mercury vapor boiler; 7, heater for liquid mercury; 8, condenser boiler; 9, superheater (steam); 10, steam turbine; 11, feed water economizer; 13, mercury turbine; 21, steam turbine condenser; 23, electric generator; 24, mercury vapor pipe.

air, gasoline, etc.), and depends solely on the temperatures t_2 and t_1. It is a common fallacy, among persons not well versed in these things, to suppose that for an engine working at a low temperature (as for example in the exploitation of the sun's radiation) some volatile substance, such as ether, would offer advantages over water. As already stated, this is not the case, the efficiency is wholly independent of the nature of the working substance, given a stated temperature of boiler and condenser.

Yet Mr. W. L. R. Emmet, working in the research laboratories of one

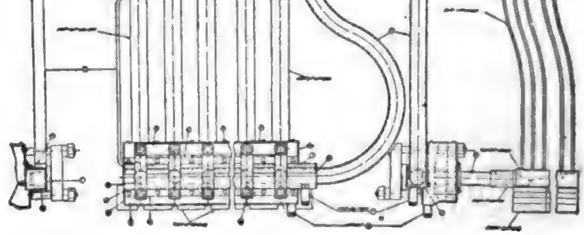


Fig. 2.—Assembly of Experimental Mercury Boiler.

of our most progressive industrial enterprises, has recently spent much time and effort developing an engine which uses the costly element mercury in place of water in the boiler. How is this to be harmonized with what was said above? The explanation is simple enough. It is true that the theoretical maximum efficiency of a steam engine working between, say, 677 degrees F. (the boiling point of mercury at atmospheric pressure) and 100 degrees F., would be exactly the same as that of an engine in which mercury vapor took the place of steam. *But,* a steam engine working at such temperatures would be impracticable, owing to the enormous pressure to which the boiler would be subjected. A mercury vapor engine can be, and has been, built to work effectively at the high temperature stated. And the higher the boiler temperature the greater the efficiency.

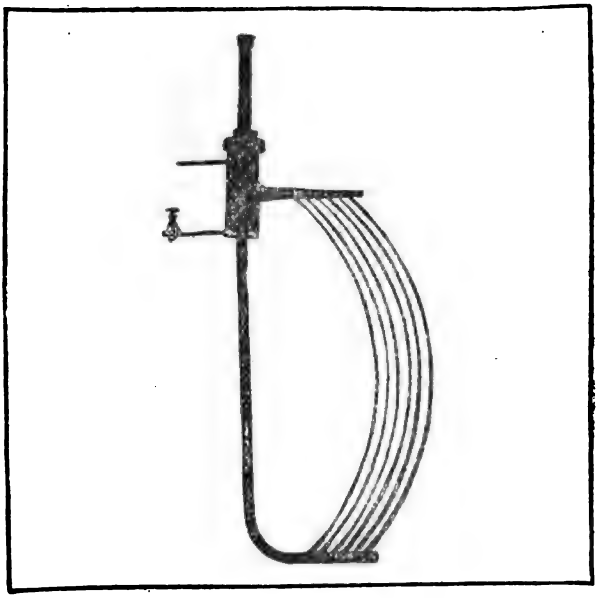


FIG. 3.—SECOND EXPERIMENTAL MERCURY BOILER TUBE.

To recapitulate, the nature of the working substance has no *direct* influence upon the efficiency of the engine. Indirectly it affects this efficiency, in so far as it determines, owing to practical considerations, the temperature at which the engine can be worked.

This, then, is one reason for Mr. Emmet's labors in perfecting a mercury vapor engine.

But there is another point to be considered. The steam engine, as commonly operated, is extremely wasteful in one particular: The combustion of coal furnishes a temperature of some 2,700 degrees F. The proper conditions are, therefore, presented for obtaining a high thermodynamic efficiency. But owing to the properties of steam a large amount of heat must be allowed to go to waste through the stack, and, though a temperature of 2,700 degrees F. is available in the furnace, actually only about 600 degrees F. (using a superheater) can, at the very best, be made use of in the cylinder. Now Mr. Emmet does not propose to work his

mercury vapor independently, but to intercalate it between the furnace and the steam boiler, so that all the advantages of the steam engine will be obtained as before, and the additional power furnished by the mercury vapor engine will be a net gain.

Whether the type of engine developed by Mr. Emmet is destined ultimately to figure prominently in industrial operations is perhaps at the present moment somewhat open to doubt. The prospect seems fairly promising, as will be seen from the description given below. In any case Mr. Emmet's engine presents many features of the very highest interest, owing to the peculiar conditions which have been faced and the ingenious manner in which the problems involved have been solved.

The method applied in his process is described by Mr. Emmet in a paper read before the American Institute of Electrical Engineers, as follows:

"Mercury is vaporized in a boiler heated by a furnace of ordinary type. From this boiler (see Fig. 1) it passes at a pressure near or not much above the atmosphere, to the nozzles of a turbine which drives a generator or other utilizer of power. From this turbine it passes to a condensing boiler where it is condensed on the outer surface of tubes which contain water, and this water is vaporized by the heat delivered, and the steam produced is used to drive other turbines or for any other purpose. This condensing boiler is preferably placed at a level above the mercury boiler, so that the condensed liquid will run back into the mercury boiler by gravity without the aid of a pump. Since the mercury vapor is much hotter than the steam, the gases will normally leave the mercury boiler at higher temperatures than they have in leaving a steam boiler. To utilize this excess heat in the gases it is proposed to convey them, first, after leaving the mercury boiler through a heater which raises the returning liquid almost to the boiling point; second, through a superheater which superheats the steam delivered by the condensing boiler; and third, through an economizer which heats the feed water for the condensing boiler and so reduces the gases to the lowest practicable flue temperature.

"By careful study and experimental development means have been devised for reducing the amount of mercury used, for effectively preventing its loss or dissipation, and for immediately detecting any failure in such prevention."

Mr. Emmet then goes on to discuss the disadvantages and the advantages of mercury for a heat engine.

"The disadvantages of mercury for such a process are: First, that it is very expensive, its cost being about 60 cents per pound. Second, that it is poisonous and is capable of pervading the atmosphere in a very finely divided state in the neighborhood of places where the vapor can escape. Third, there are certain difficulties in confining both the vapor and the liquid, although these, with proper methods, are not serious.

"Mercury's advantages as a thermodynamic fluid for the purpose desired are many. First, its boiling points at desired pressures are convenient. Second, its high specific gravity makes possible the use of gravity feed, sealing of valve stems, etc., and centrifugal sealing of turbine packings. Third, it is completely neutral, at the temperatures used, to air, water, iron, and such organic substances as it may come in contact with. Fourth, it carries nothing in solution which can adhere to or affect heating surfaces, consequently the interior of boiler is always perfectly clean. Fifth, its vapor density is so high that it gives a very low spouting velocity, and consequently a very simple type of turbine can be used. Sixth, it does not wet the surface of turbine blades and consequently gives apparently no erosion. Seventh, its volume at convenient condensing temperatures is such that it can be used in turbines without excessive bucket heights. One of the greatest limits of design in steam turbines is

the large area required for the efficient discharge of the low-pressure steam. With mercury vapor this difficulty does not exist. Eighth, delivering its heat at the temperature and in the manner which it does, the condensing boiler in which this heat is used to make steam is very small and simple as compared with a steam boiler. Steam boilers transmit an average of about 6 watts per square inch with an average temperature difference of about 1,100 degrees F. A surface condenser transmits 18 watts per square inch with 20 degrees F. temperature difference. The mercury boiler is about equivalent in dimensions to a surface condenser, and since there is no high temperature involved, there will be no possibility of scaling or burning. High temperature, unequal distribution of heat, and the necessity for large heating surface constitute the principal difficulties of boiler construction. All of these are overcome in this method of making steam."

As regards the economy to be gained by his process, Mr. Emmet remarks:

"Before entering into the details of experiments or of the methods by which it is hoped to accomplish these results, it may be well to state the degrees of economy which should be accomplished if this development succeeds. Assuming heat deliveries to surfaces exposed equal to those in steam boilers under equivalent conditions of temperature difference, gas velocity and radiation, and assuming a turbine efficiency equal to that of steam under equivalent velocity conditions, the calculation shows that in an efficient modern power station the same amount of steam can be delivered to the turbines at the same superheat, thus giving the same turbine output, and that in addition about 66 per cent. of the power so delivered can be delivered by mercury turbines, the fuel required being only about 15 per cent. greater than that which would be used with the steam alone. Thus the gain in capacity of an existing station would be approximately 66 per cent. and the gain of output per pound of fuel would be about 44 per cent. This calculation is based upon a mercury vapor pressure 10 pounds above the atmosphere and a vacuum of 28.5 inches at the steam turbine outlet.

"About 10 pounds of mercury would be evaporated for each pound of steam produced, the steam pressure being about 175 pounds gage, superheat 150 degrees F., and the final temperature after gas leaves economizer being about 300 degrees F. The vacuum in both steam and mercury turbines can be maintained by the same air pump, means being employed to separate all mercury vapor from the air in a suitable cooler.

"Experimental data indicate that not more than $10 worth of liquid mercury per kilowatt output of mercury turbine will be required for such a process, and it is probable that with suitable arrangements this amount can be considerably reduced. The general application of such a process would require immense quantities of mercury, but inquiry has indicated that the sources of supply are such that the largest conceivable demand for such a purpose would not permanently increase the price."

Mr. Emmet then proceeds to describe his mercury boiler:

"With a view to developing a design for a mercury boiler which would produce the vapor with the practicable methods of firing without destructive temperatures in the steel and with a small total amount of mercury in use, it has been necessary to do much experimenting to determine the behavior of mercury under boiler conditions. The construction of the boiler as finally developed is shown by Fig. 2, and the experimental unit used to determine its characteristics is shown by Fig. 3.

"This boiler is made up of a number of heating units, each consisting of an upper and a lower header which are connected together by curved flattened tubes. The flattening is to reduce the space which must be filled with liquid mercury without diminishing the surface. The curvature is to prevent mechanical strains through unequal expansion caused by irregular

heating. These flattened tubes are connected into the headers by acetylene-welded joints. The tubes are first welded from the inside into channel-shaped pieces; these channels with the set of tubes connecting them are then annealed so as to release all strains incident to the welding. After annealing, they are tested with high-pressure air, suitable clamps being used to confine the air in the channels. These channel pieces are then welded to steel headers so that the whole unit becomes perfectly tight and capable of standing a high pressure. The headers of these units terminate in taper nozzles, which fit into taper holes in the bus header at the bottom and into a vapor chest at the top. A curved liquid duct connects the vapor chest to the bottom header at the hot end so that the heating units which are exposed to the greatest heat receive the most direct supply of liquid. In these hot units a larger internal space will be allowed than in the units which occupy the cold part of the boiler, so that the colder part will not carry an unnecessary amount of liquid.

"The arrangements of the turbine have been improved as the result of experience. The first turbine arranged was made from an old experimental steam turbine, and had many joints which were difficult to keep vacuum-tight. The new experimental turbine has only one joint, arranged in a manner which has been experimented with and which will avoid leakage.

"The condensing boiler used in the experiment now in preparation is made from a standard high-pressure feed heater having a water space at the top and bottom connected by tubes in the manner customarily used in such devices. This boiler has apparently worked satisfactorily and produced steam from mercury vapor in the manner expected. It is not, however, considered a suitable design for the purpose, since the temperature differences impose a strain on the expanded tube sheets. It is thought that tubes attached at one end with concentric circulation after the manner of the Nicholaus boiler will afford the most satisfactory method for such condensers. Since no violent temperature differences will exist, it is believed that these condensing boilers can be made practically free from deterioration and entirely free from leakage."—"Scientific American."

CRACKED AND SEIZED PISTONS ON DIESEL ENGINES.

By Geo. E. Windeler.

(*Read before the Diesel Engine Users' Association.*)

In approaching the subject which we have before us today I have in mind that there has been a very considerable abount of research work carried out in different parts of the world in connection with the "growth" of cast iron and the difficulties with cast iron when subjected alternately to high and low temperatures, and therefore any remarks that I make on the subject are entirely based on the investigations which we have carried out at the works of the firm with whom I am associated.

Naturally there are many points in connection with the investigations which I am not at liberty to disclose, but I will endeavor to place before you the procedure that has been adopted and the results that have been obtained step by step, coupled with the modifications that have been made in design to meet the decisions that have been come to at the different stages of the investigations.

I am exhibiting a number of diagrams illustrating the different stages in the design of pistons we have used on various sizes of engines, and also a diagram, Fig. 6, showing the general arrangement of the combustion space, with the valves and piston in position, to illustrate my remarks in regard to the transmission of heat through water spaces.

In dealing with the internal-combustion engine as a heat motor, the points of difference between it and the steam engine are considerable. In the steam engine, to obtain the most economical results, everything possible is done to keep the temperatures of the cylinder liner, piston and ports for the incoming steam, and all surrounding spaces, as near as possible to the temperature of the live steam. To obtain the most efficient results steam jacketing has been resorted to, latterly superheated steam, and the latest innovation has been the adoption of the Uniflow engine, which admits steam by a set of ports which are placed at one end of the cylinder, and allows the exhaust steam to pass away through another set of ports situated either at the other end, or at the middle of the cylinder, according to the type of engine.

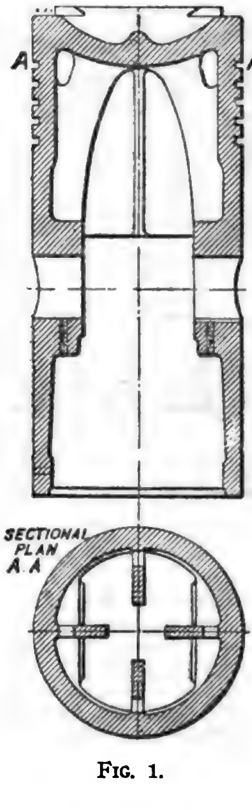


FIG. 1.

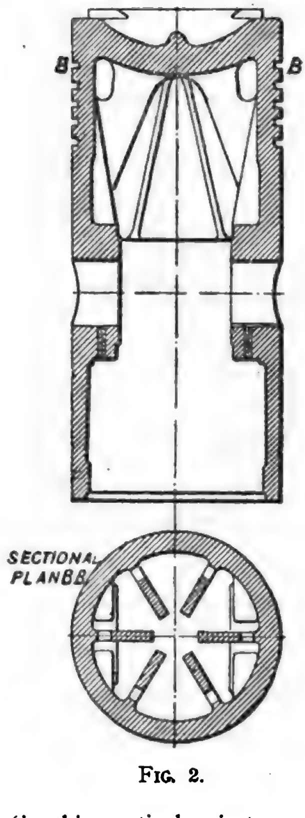


FIG. 2.

In the internal-combustion engine (in this particular instance I will deal specially with the Diesel engine) everything possible is done to keep down the temperature of the cylinder liner, cylinder cover or breech end, exhaust valves and exhaust passages which are in contact with the fuel or gases. In some types the pistons are water or oil cooled. In a well-designed Diesel engine 40 per cent. of the heat units admitted to the combustion space is absorbed in useful work. Of the remainder, 25 per cent. passes to the exhaust and 35 per cent. passes away to the circulating water and is lost in radiation, so that almost as much heat has to be absorbed by the cooling water as is developed in useful work; therefore

special attention has to be paid to water-cooling arrangements to ensure that the circulation is thorough and that no air or steam pocketing is likely to occur, and special attention should be paid to the arrangements to prevent short circuiting in the water circulation, otherwise certain parts are liable to become overheated locally.

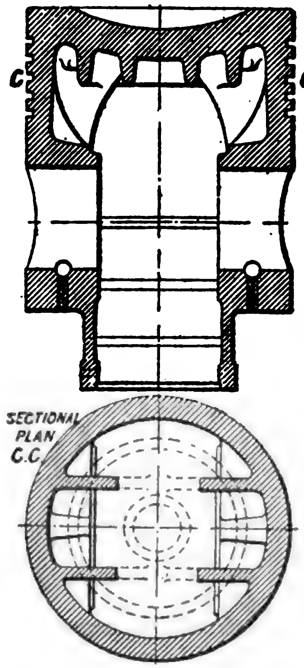


FIG. 3.

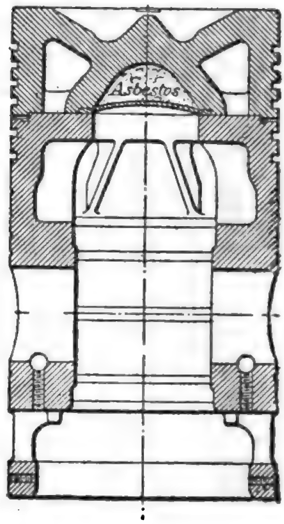


FIG. 4.

In the course of the investigations it was found that the major portion of the defects that had arisen on cylinder covers and pistons were due to serious deposits having been allowed to form in the water spaces of the cylinder covers and on the cylinder liners, and a good deal of time was spent in investigating the cause of such deposits having formed, and ultimately we came to the conclusion that such deposits were not formed on the liners and cylinder walls during the time the engine was working but after the engine was shut down. Naturally, the piston, cylinder cover, valves and such parts as are in contact with the combustion space must have a large number of heat units stored in them, and when the engine is shut down these heat units are absorbed by the water which remains in the jacket, the quantity of which is comparatively small, and a very considerable rise in temperature takes place; in fact, sufficiently high in the majority of cases to throw down the salts which were held in solution in the water. We were so sure that the theory was correct (which, as far as we knew, was quite a new interpretation of the difficulty) that we wrote to the various users of our engines suggesting to them, where the circulating water was known to be of a considerable hardness, that they should run water through the water spaces for some time after the engine was shut down—with the most happy results. This

information, naturally, became public, and most Diesel engine users have now adopted this system to their advantage.

I have personally inspected a cylinder cover which contained a water space about 6 inches deep and found a deposit of as much as 5 inches thick in the water space. Naturally, the cover had cracked. The aforesaid suggestion was adopted in this particular station, and the results have been very satisfactory indeed.

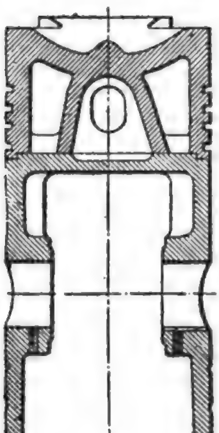


FIG. 5.

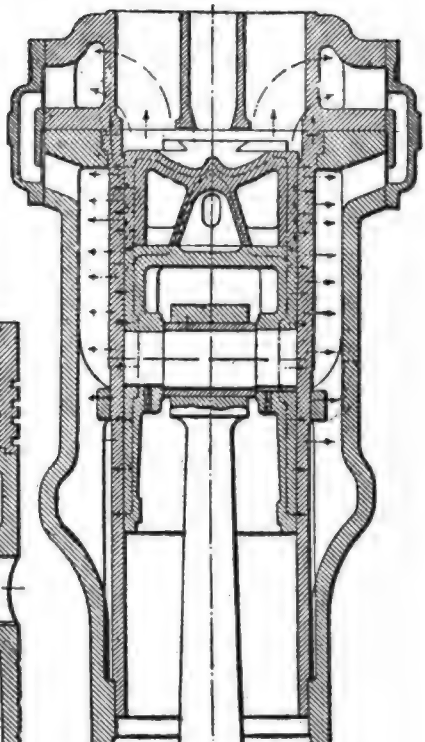


FIG. 6.

In an internal-combustion engine without a water-cooled piston the heat units that have been absorbed by the piston crown must, out of necessity, be dispersed from the crown down the body of the piston, and passed away through the liner to the water space by the contact that the piston forms with the liner. It is therefore necessary to study carefully the design of the piston head to obtain the best results. At the same time the piston must be designed in such a way that the load that is applied to the crown shall be transmitted to the crosshead pin which attaches to the connecting-rod, without causing any distortion to the piston, which has to maintain its roundness to enable it to work without trouble in an accurately-bored cylinder.

In the small internal-combustion engine such as is used on automobiles and air craft, there is quite a difference of opinion as to whether cast iron

or oil-tempered steel is the most suitable material to use for pistons. One would almost imagine that the oil-tempered steel piston would be the most suitable for such machines, where high power is developed in comparatively small cylinders, and at a high rate of revolution; but for large engines such as we have under discussion it is necessary to have a fairly heavy section of material to enable the heat units to pass away in the time available, so as to prevent undue heating of that part of the piston which is in contact with the gases.

In diagram No. 1 is shown a piston with the usual dished top, and which is supported by four radial ribs, two ribs being carried down to the piston-pin boss so as to form a more or less rigid connection between the piston crown and the piston pin, to enable the forces to be transmitted direct with the minimum amount of distortion. It will be noted that this piston is comparatively thin in the crown and in the body, and when fitted to a number of engines, was found to work admirably; but ultimately distortions appeared between the ribs, and it was deemed advisable to modify the design to that shown on Fig. 2, in which six radial ribs are carried well down the body of the piston from the crown, two being carried to the piston-pin boss, with a thicker rib behind the main rib to assist in the transmission of the load. The piston crown was made slightly thicker, and generally this design has been satisfactory, but at times cracks have developed in the piston top, sometimes the cracks having taken a circular form round the base of the center cone which is formed in the piston. Such cracks, however, are not of serious import, and pistons which have cracked in this manner have been in use for twelve or fifteen months without any trouble, a slight blowing past taking place when starting up, but which ceases after the piston top has become heated, evidently the expansion closing up the crack. But a more serious state of affairs exists when the piston cracks either radially or diametrically, as it is liable to cause seizures in the cylinder owing to the piston losing its rigidity, and consequently its symmetry. It is desirable under such circumstances to immediately shut down and remove the piston.

The diagrams 1 and 2 refer to a piston 12 inches in diameter; diagram 3 shows a piston of 20 inches diameter, the design of which was similar to the 12-inch, but the piston crown was supported by a number of circular ribs projecting into the interior of the piston. Radial ribs were also carried down from the crown of the piston to the piston-pin boss. This piston at first appeared to be quite satisfactory, but it was noticed on examining a number of castings which had been unmachined that slight cracks had taken place across the tips of these ribs, and on examination it was found these cracks had been set up by the cooling of the casting.

A number of such castings were broken up and found to contain many flaws. It was therefore decided to cast the piston in two portions, and diagram 4 illustrates the type of piston which is now generally adopted by the firm. The construction of the upper portion of the piston is secured to the main body of the piston by six or eight square-necked studs; the object of the square-necked stud, which fits into a square hole in the body of the piston, is to ensure that the stud itself will not come out or get slack when the engine is at work.

The design of this piston top is somewhat novel, insomuch that there are no ribs embodied in the design, the support to the crown of the piston top being obtained by throwing out a conical cylindrical chamber which also has the advantage that it forms another lane by which heat units can reach the body of the piston for transmission to the water spaces. It also permits of uniform expansion. The upper portion of the piston is carefully bedded on to the lower portion, and care has been taken to have sufficient contact area to allow for rapid heat transmission. The internal cone thus formed in the head is filled with asbestos to prevent the heat being thrown down from the crown of the piston on to the connecting-

rod bearing. This design has several advantages, one being that different quantities of material may be used for the upper portion of the piston and the lower portion. The other is, that should any cracks develop, it is a comparatively simple and inexpensive matter to replace the top. As will be readily understood, the lower portion, which acts as a crosshead, must necessarily be of a quality of iron which is hard and which will attain a high polish when running, and not wear readily; the upper portion should be of a material capable of resisting heat stresses.

Diagram 5 shows a similar design as adopted on the 12-inch pistons. When these modifications were carried out, a number of experiments and analyses were made by our laboratory with the different qualities of material used, and careful records were taken of such, and when any defects developed we were able to trace the particular quality of material that was used. By this comparative method we were able to arrive at a more or less definite conclusion as regards the quality of material which best suited the purpose. Ultimate experiments proved that cast iron containing high percentages of phosphorus were most unreliable, and further investigations proved that the eliminations of phosphorus would, to a large extent, remove the difficulty of cracks developing. Great care is taken by us in controlling the mixtures that are used, and each batch of pistons and cylinder covers, liners, etc., is carefully analyzed, and records taken by our laboratory so as to ensure consistency of the material as desired.

Reverting to the question of seizures of pistons, we have formed the opinion that almost invariably these seizures are due to heating of the piston pin and its bearing, and from our investigations we have found that such seizures take place almost immediately after starting up. The opinion we have formed is that, when an engine has been set to work and shut down, the heat stored in the piston body is thrown down and tends to evaporate the oil in the top end bearing, with the result that when the engine starts up again, this particular bearing is not amply supplied with lubricating oil, and seizures take place. Again, running the circulating water through the engine after shutting down tends to minimize this difficulty.

In the enclosed type of engine, where forced lubrication is used, this difficulty is to a large extent removed by the fitting of a hand-lubricating pump, by means of which lubricating oil under pressure can be pumped to all the bearings, including the piston-pin bearing, whilst the engine is being barred round into its starting position before being started up.

As proof of the contention that the heating of this bearing is generally the cause of such seizures, it may be mentioned that the abrasions generally show seizure to have taken place at four points of the circumference of the piston and liner, and if careful note is taken of the position of these abrasions, it will be found that they occur at the point where the piston-pin bosses join on to the thinner section of the body of the piston. At this point there is a comparatively sudden reduction in the section of material, allowing one portion of the piston to be extremely rigid and the other portion more or less flexible; when seizures take place the piston pin becomes overheated, and by reason of its friction fit in the piston body, plus its greater coefficient of expansion, causes an extension to take place, which forces the piston out of shape, flattens the crosshead portion of the piston body, and causes distortion to occur at the four points referred to. If the engine is not shut down quickly, other seizures are liable to occur, but the main seizures will almost invariably be found to occur at these four points.

When piston seizures do occur it is desirable that the connecting-rod bolts should be carefully examined, as extension of the length of the bolt is likely to take place, and if examination is not made, ultimately fractures of the bolts are likely to occur, with most disastrous results.—"Page's Engineering Weekly."

CONTROL AND PROTECTION OF ELECTRIC SYSTEMS.*

By Charles P. Steinmetz.†

When the first commercial electric circuit issued from a station the problem of control and of protection arose. It was a simple problem at first—an ammeter and voltmeter to measure current and voltage; a knife-blade switch to send the current into the desired path or withdraw it; the fuse to open the circuit in emergencies, and if the wires became crossed and the fuse and switch failed, the generator and engines stopped, and not much harm was done. With the extension of the circuits into the suburbs some lightning troubles were felt, which led to the introduction of lightning arresters.

Since those days, less than a generation ago, enormous changes have taken place, and the electric systems have increased in size, in voltage and in extension. Where 100-H.P. machines were large once, now turbo-alternators of over 40,000-H.P. are in commercial operation. The steam engine has made room for the steam turbine, and the steam turbine does not stop when the wires are crossed and a short-circuit occurs; the momentum of the turbine discs, revolving at velocities of 300 to 400 miles per hour, can supply ample energy for the destruction of any part of the system. Attempts to open such circuits by the knife-blade switch of old would lead to the destruction of the switch, and probably its operator.

Instead of small machines operating separately on independent circuits, huge generators now feed in parallel into the system of busbars, on which is concentrated all the power of the station or the group of stations tied together. Numerous stations and systems of interconnected stations of 100,000 to a quarter-million horsepower and over are in operation, and the half-million horsepower mark has been reached.

Anywhere on the busbars of the station or in the feeders near the station there is available, destructively in case of an accident, as a short-circuit, not only the entire power of the station, but the far greater power which the station generators can give momentarily. Short-circuit currents of forty to fifty times normal full-load current may momentarily flow from some turbo-alternators, representing ten and more times full-load power. Such a station, or group of closely interconnected stations, of half a million horsepower full-load capacity, may momentarily send into a short circuit at the busbars over five million horsepower. This is the power of Niagara, which is estimated variously at from 5,000,000 to 15,000,000 H.P. It is obvious that no switch or circuit breaker can be built to safely relieve such power.

With half a million horsepower station capacity, a momentary overload capacity ten times as high, assuming that we could build a circuit breaker to open this short-circuit power as quickly as in three to four cycles, or one-eighth second, would require dissipating in the circuit breaker the energy of over 200,000,000 foot-pounds—the destructive energy of 1,000 tons dropping from a height of 100 feet, that of a projectile weighing 2,000 pounds leaving the cannon at a velocity of 2,500 feet per second, or the destructive energy of two heavy railway trains, going at sixty miles per hour, and meeting headon.

Equally great has been the increase of voltage. Where once 2,000 volts was high, in circuits of a few miles in length, now circuits of over two hundreds miles are in operation at 100,000 to 150,000 volts. Current at such voltages jumps toward an object over a foot distant, and will maintain arcs of practically unlimited distance; that is, with 100,000 volts and almost unlimited power back of it, an arc can extend for several hun-

* From a paper presented at a joint meeting of the Philadelphia Section, A. I. E. E., and the Franklin Institute and printed in the July "Journal" of the Institute.
† Chief consulting engineer, General Electric Co.

dred feet. Thus no simple switch will open a circuit at such voltages under power.

Lightning protection also has become a far larger problem than in the small circuits of old. But far greater than the energy of any lightning stroke is the energy stored as magnetic field surrounding the conductors and as dielectric field radiating from the conductors of these big transmission systems, and, if this internal energy of the system is set surging, its effects are far more destructive than those of lightning. The effects may not be merely momentary, as those of lightning, but continual, as machine energy keeps replacing the stored internal energy which causes the destructive surge. The foremost problem of control of electric systems thus is that of controlling enormous powers; the foremost problem of protection is that against self-destruction by its own power.

Current and voltage have grown beyond the values for which instruments can be built, and current transformers and voltmeter transformers are interposed between the circuit and the instruments measuring it. With the general introduction of parallel operation, power-factor indicators are required to insure the division of load without excessive waste currents, and frequency indicators and synchronizing devices to safely connect machines into the system.

With hundreds of feeders radiating from the generating station the office of the load dispatcher has become essential, and the necessity of keeping exact records of all operations and of all accidents and incidents is of the greatest importance. Automatic recording devices thus have been developed, as the multi-recorder, to record within fractional seconds all important events—opening and closing of switches, starting and stopping of generators, surges, lightning disturbances, etc. Such automatic devices afford a valuable check on the operating staff, but more important still is their record in emergencies. Where a number of things happen almost at once and the attention of the operator is distracted from accurate observation by the necessity of action, the record thus could be made only afterwards from memory. It is just in such abnormal conditions where the most complete and accurate record is of greatest importance, to enable the engineers to determine with certainty what happened and why it happened, so as to take steps to guard against its recurrence.

Oil circuit-breakers have been developed, which can safely and without disturbance close and open feeder circuits of over 10,000 H.P. and generator circuits of 40,000 H.P. In these the circuit is opened under oil so that at the instant of opening the current is extinguished at the end of a half-wave by the rapid expansion and chilling of the oil vapor produced by the opening arc; this at first is under high compression, due to the momentum of the oil which has to be set in motion.

POWER-LIMITING REACTANCES.

The possibility of self-destruction by the power let loose under short circuit was solved by the development of the power-limiting reactances. In the generator leads, between generators and busbars, are inserted reactances capable of withstanding enormous overloads, but of a size sufficiently small not to interfere with the normal flow of power at full load or any overload that the generator may be called upon to carry. They are large enough, however, to materially limit the generator current and power on short circuit. Usually the generator reactances limit the momentary short-circuit current to about ten to twelve times full-load current; that is, the momentary short-circuit power to about two and one-half times full-load power. This solved the problem for medium-sized stations.

However, even with generator reactances, with the increasing size of station, the power that may be let loose under short circuit becomes large beyond control, and busbar reactances were next introduced. By such reactances in the busbars and in the tie feeders between the stations, the

system is divided into sections of about 60,000 H.P. each. A short circuit then can seriously involve one busbar section only.

With hundreds of feeders radiating from the busbars, the probability of a short circuit in the feeders is far greater than in the busbars, and a material advantage, therefore, is given by feeder reactances.

By the development of generator reactances, busbar reactances, and feeder reactance, the problem of the power control of large systems for protection against self-destruction by short circuit has been solved, and unlimited extension of systems without any increase of danger has been made possible; and experience has shown that after the introduction of such power-limiting reactances dead short circuits have occurred at the busbars of very large systems without even interfering with the operation of most of the synchronous apparatus on the system.

GROUNDS AND SHORT CIRCUITS BETWEEN PHASES.

The two main sources of trouble in lines and cables are grounds and short circuits between phases. In transmission lines a ground on one phase is the most frequent trouble, and short circuits are rare except in lines in which the design is faulty or reliability has been sacrificed to cheapness. A short circuit is far more serious than a ground, as in the former the current is limited only by the generator capacity, while with a ground the current has no return unless the neutral is grounded, and then over the resistance of the neutral. In a well-designed transmission line a short circuit usually occurs only as the result of two simultaneous grounds. A ground on one conductor, however, raises the voltage to ground of the other two phases, from the Y voltage to the delta voltage of the system, and thereby increases the strain on the insulation ot the other two phases. It thus introduces the danger of a second ground, causing a short circuit, or requires higher insulation.

This has led to two methods of operation of transmission systems. In the first, the neutral of the transformers is grounded, frequently through a resistance, where the resistance of the ground is not high enough to limit the current. Then a ground on one phase is a partial short circuit to the neutral, causing a large current to flow, and thereby opening the circuit breakers before the ground has developed into a short circuit. However, this method, the grounded Y system, means a shutdown at every ground, every flashover of an insulator by lightning, etc. In the second method the neutral is not grounded, but the insulation of the circuit is good enough to safely stand the increased strain put on it by a ground on one phase, and by means of an arcing ground suppressor, etc., care is taken not to continue the arcing ground, leading to high-frequency disturbances, but convert it into a metallic ground. In this case the "isolated delta" system can be maintained in service, even if one phase grounds, until arrangements are made to take care of the load or the fault is found and remedied. However, the cost of line construction is higher, because of the better insulation required. The relation between grounded Y and isolated delta is one of cheapness versus reliability and continuity of operation.

PROTECTING UNDERGROUND CABLES.

Different are the conditions in underground-cable systems. In a cable the three conductors are so close together that a ground on one conductor quickly reaches the others and becomes a short circuit. A grounded cable, therefore, cannot be kept in service, but has to be cut out promptly. In these systems it is customary to ground the neutral through a resistance sufficiently low, in case of a ground on one conductor of a cable, to allow enough current to flow to open the circuit breaker and cut off the cable, but not sufficient to give a severe shock to the system. Or, where grounding of the neutral is considered undesirable, an arrangement of relays is made to give the same effect. With underground cables such cutting off of a disabled feeder does not interfere with the continuity of

service, as a number of feeder cables are always used in multiple for every important substation.

However, the problem of cutting off a disabled feeder by the operation of the circuit breaker, owing to the large current taken by the grounded feeder, is not so simple. Therefore, so-called "inverse time-limit" circuit breakers are generally used; that is, circuit breakers in which the time limit of their operation decreases with increasing overload. Such circuit breakers would first cut off the cable carrying the greatest excess current—that is, the faulty cable—and then those of less excess current; but, as the excess current stops with the cutting off of the faulty cable at both ends, other cables should not be interfered with. However, the inverse time-limit circuit breaker necessarily must be practically instantaneous under short circuit, and therefore, while the time limit discriminates between 100 per cent. or 200 per cent. or 300 per cent. overload, it cannot discriminate between short circuits of different magnitude.

Thus devices become necessary for selecting a disabled feeder and cutting it out without cutting off its parallel feeders or the tie feeders to the substation served by the faulty feeder, regardless of what excess currents these may carry. This is a problem that has not yet been completely solved.

In general, in high-power systems of high standard of reliability the radial system of substation supply is used; that is, each substation is fed by a separate set of cables, and the substations are not interconnected into a network by a system of tie feeders. This radial system, however, is less economical in feeder copper than the interconnected network, since the radial system requires for each substation a feeder capacity equal to the maximum power demand of the individual substation, while in the network, by cross-feeding between the substations, the feeder capacity is reduced to that required by the average maximum demand of the substations. Because of the economic disadvantage of the radial system an effective selective feeder relay that could be relied upon to pick out the faulty feeder and no other would offer material advantages.

Such a selective device is afforded by the use of pilot cables. Each cable or feeder is duplicated by a smaller low-voltage three-phase cable, which joins the secondaries of current transformers connected into the two ends of the main cable. If the main cable is undamaged the same current comes out of it as flows into it, and the connections to the pilot cable are such that in this case the secondary currents would be in opposition; that is, neutralize each other. If, however, the main cable grounds, current flows into it from both sides, the secondary currents in the pilot cable then add, and the current flowing in the pilot cable operates the relay which opens the circuit breaker. This arrangement is very perfect in operation, and is capable of cutting out the damaged cable without interfering with any other. but it has the disadvantage of doubling the number of cables required in the system. While the pilot cables are small and of low voltage, they occupy room in the underground ducts; hence, this method of control is little used in this country.

Another method is that of the split-conductor cable. Every cable conductor is made of two parts, of which the one surrounds the other concentrically, with some insulation between them. Normally there is no potential difference between the inner and the outer half of the conductor, as they are connected with each other at the ends of the cable. If, however, a ground occurs on the cable, this ground can at first reach only the outer half of the conductor, and a potential difference and current appears between the inner and the outer half of the conductor and operates the circuit breakers through a relay connected between the halves of the conductor, at either end of the cable. This method also works very satisfactorily, but has the same economic disadvantage, though to a lesser degree than the method of pilot cables, in that the split-conductor cable is materially larger and more expensive.

The usual method of taking care of the problem, at least in most cases, is by the so-called reverse-power relay, also wrongly called "reverse-current relay." Such a reverse-power relay operates perfectly so long as there is any voltage for the reverse current to act upon. If, however, a short circuit occurs at or close to the substation, the voltage vanishes, and with it the reverse-power relay loses its pull. To guard against this the installation of reactances is recommended between cables and substations to give a sufficient voltage drop to operate the relay. However, this is an additional complication.

The reverse-power relay is not adapted to guard the feeders between stations, as in these the current reverses in direction with the change of the distribution of load between the substations. Thus the reverse-power relay does not make the operation of interconnected networks of substations possible, but in the radial system of operation it is the only device which is generally available economically, and it is very satisfactory, with the exception that it cannot operate where there is no voltage left.

LIGHTNING INTERFERENCE.

Interference by lightning with high-potential transmission lines, has rather decreased with increasing line voltage. In 100,000-volt lines the insulators are tested for one minute at 200,000 to 250,000 volts and stand momentarily for the very short time of lightning, over half a million volts. Thus it is rare that lightning flashes over or punctures the suspension insulators of our very high-voltage transmission systems. A flashover, with the grounded Y system, shuts down the circuit, often without any damage, while with the isolated delta system it may not even shut down the circuit, but is taken care of by the protective device against flashovers—the arcing ground suppressor in the station. Most lightning voltages incapable of destroying the line insulation run along the line until their energy is dissipated or they reach a station, and there they often do serious damage. The most important problem of lightning protection thus has become the rapid damping out of line disturbances caused by lightning, so as to make them harmless before they reach the station. The most effective method has been the overhead ground wire. By its screenings effect it lowers the voltage which lightning can induce in the line, but far more important is its powerful damping effect on the line disturbance—the traveling wave caused by lightning which runs toward the station.

In considering the protection of modern electrical systems it must be realized that the various sources and kinds of interference or danger require correspondingly different protective devices. It would be just as unreasonable to expect a standard type of "lightning arrester" to protect an electric system against all possible troubles as it would be to call for a single-standard "safety device" which would protect a railway train against all possible dangers, from a broken rail or a washout to a collision or a boiler explosion.—"Power."

FIRST AMERICAN REVOLVING FLOATING CRANE.*

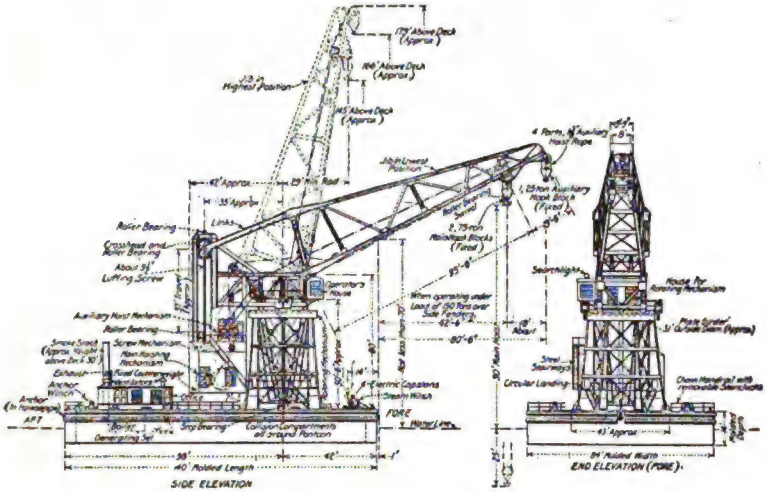
HIGH-CAPACITY LIFTING APPARATUS FOR NAVY YARD.

A. F. CASE—"ENGINEERING NEWS."

For the Norfolk Navy Yard a 150-ton capacity floating crane, illustrated herewith, is to be built by the Wellman-Seaver-Morgan Company, the first of its kind constructed in this country. It will weigh when built 2,500 tons and cost $350,940.

* cf. "The Engineering Magazine," July, 1915; p. 584.

It consists of a steel tower on a steel pontoon with a pivot frame carrying the operating jib revolving in the tower. The crane will lift 150 tons at a maximum reach of 62½ feet over sides or one end. This load will be carried on two separate hooks some ways back from the end of the jib, and an auxiliary hook on a trolley capable of operation from the end of the jib will be capable of hoisting from a point 25 feet below the water to 130 feet above the water at 62½ feet beyond the fender. The jib can be luffed under load from maximum reach to a minimum of 29 feet.



ELEVATIONS SHOWING ESSENTIAL FEATURES OF THE CRANE.

The crane will be electrically operated, all the machinery being carried in a machinery house on the rotating structure. Power is obtained from a steam-operated generating set on the pontoon. The pontoon will be 85 feet wide, 140 feet long and 15 feet deep.

The motors are all of one size, provided with magnetic-switch control, the master controllers being situated in the operator's cab. The electric current will be supplied at 250 volts by a non-condensing compound engine driving a 150-kilowatt direct-current generator.

REDUCTION GEARS ON THE *PENNSYLVANIA*.

CRUISING TURBINES FITTED WITH WESTINGHOUSE HYDRAULIC FLOATING-FRAME REDUCING GEARS, DESIGNED TO TRANSMIT 1,600 HORSEPOWER.

The general features of the machinery for the United States battleship *Pennsylvania* have been briefly discussed in the engineering press. Now that the cruising gears have been built, some additional information regarding their construction will be of more than passing interest.

The *Pennsylvania* has four shafts and propellers. Each inboard shaft is driven by a high-pressure turbine placed in the forward engine room and has an independent backing turbine located in an engine room immediately aft of the forward one. Each outboard shaft is connected to two cruising turbines, located in the forward engine room, and also to the

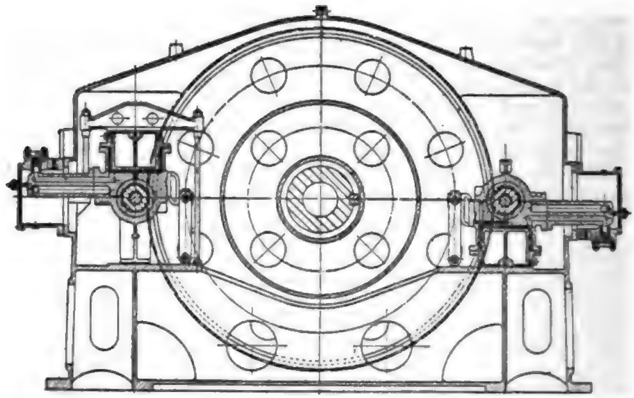


FIG. 2.—END ELEVATION.

main low-pressure turbine, with its ahead and astern elements placed in the after engine room.

At cruising speeds steam from one cruising turbine passes to the other cruising turbine, then to the main high-pressure turbine driving the inboard shaft, and finally to the main low-pressure turbine on the outboard shaft. At high speed both of the cruising turbines are disengaged and the steam enters the main high-pressure turbine and passes directly to the main low-pressure turbine.

This arrangement of propelling machinery, while adding to the number of units, has a distinct advantage in steam consumption over the earlier systems, in which the main turbines are used for cruising purposes. As about 90 per cent. of the total voyages of a battleship are made at cruising speed, the importance of obtaining the highest possible economy at this speed is apparent.

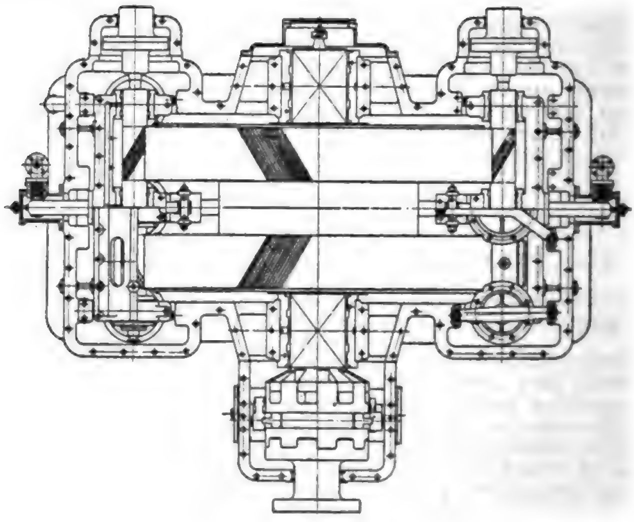


FIG. 3.—PLAN.

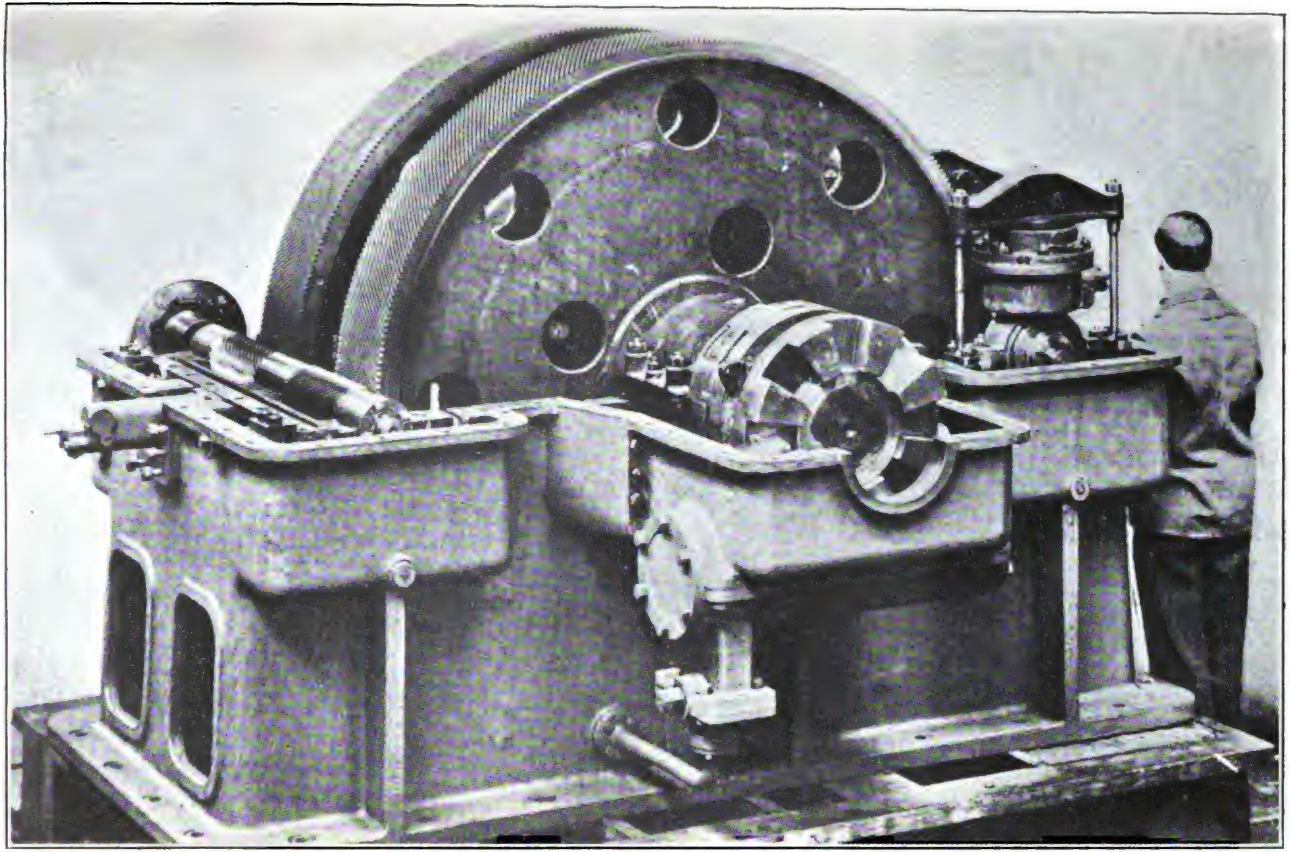


FIG. 1.—WESTINGHOUSE REDUCTION GEAR FOR CRUISING TURBINES OF BATTLESHIP "PENNSYLVANIA." SPEED REDUCTION, 1,800 TO 120; HORSEPOWER TRANSMITTED, 1,600; WEIGHT, 23,000 POUNDS.

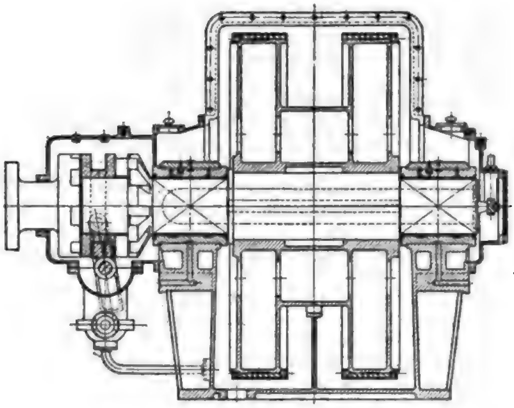


FIG. 4.—SIDE ELEVATION.

In order to get this better economy the two high-speed cruising turbines are connected to the outboard propeller through the medium of a two-pinion Westinghouse hydraulic floating-frame reduction gear. The details of these gears are clearly shown by the end elevation, Fig. 2; the plan view, Fig. 3, and the side elevation, Fig. 4. The clutch coupling on the main gear shaft serves to disconnect the cruising turbines when the vessel is traveling at full speed.

The general construction is the same as used on all high-speed, high-power gears built by the Westinghouse Company, the distinctive feature of these gears being the use of the floating frame, which insures uniform tooth pressures under all load conditions. The practical value of the floating frame has been thoroughly demonstrated, some of the first gears

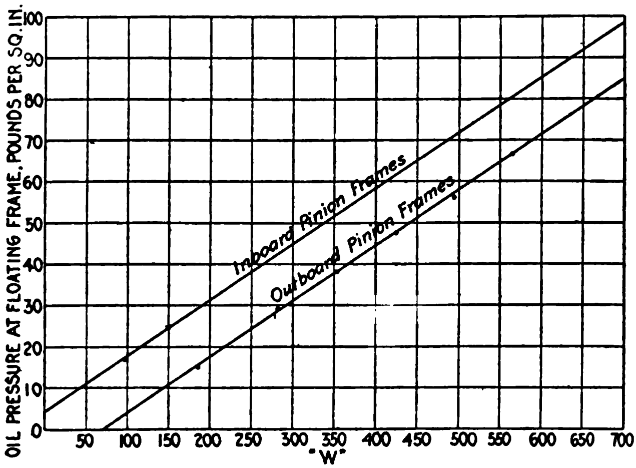


FIG. 5.—FLOATING FRAME CALIBRATION CURVES U. S. S. "PENNSYLVANIA" CRUISING REDUCTION GEARS. B.H.P. = W × R.P.M. OF GEAR WHEEL × 0.0118.

built having been in continuous service over four years. The teeth of these gears, it is claimed, show no wear that can be detected, and the uniformity of the pressure distribution is very marked, as the whole length of the gear-tooth face is well polished.

An additional advantage gained by using the floating frame is the ease with which the horsepower transmitted may be determined. Experience has shown that in a high-speed bearing, such as employed here, the oil pressure developed by the bearing is directly proportional to the power transmitted. It is only necessary, therefore, to connect the oil pipes from the floating frame to a pressure gage and obtain the speed by means of a suitable tachometer. Obviously, both of these instruments may be of the recording type, so a continuous record of the shaft horsepower developed may be obtained.

Calibration curves for these particular gears are shown by Fig. 5. The difference in oil pressures on the inboard and outboard frames is due to the fact that in one case the floating frame is above and in the other below the pinion bearings. Supposing the oil pressure on the inboard side is sixty pounds and on the outboard side forty pounds; from the calibration curves it will be seen that the corresponding values of W are 415 and 367, respectively. Multiplying the sum of these, or 782, by the revolutions per minute of gear wheel and by the constant 0.0118, the shaft horsepower is readily obtained.

The general construction of these gears and their size are clearly shown in Fig. 1. Each gear is designed to transmit sixteen hundred shaft horsepower, with a speed reduction from eighteen hundred to one hundred and twenty revolutions per minute. The weight of each gear exclusive of the clutch is twenty-three thousand pounds.—"International Marine Engineering."

NAVAL VESSELS.

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1915.		Percentage of completion Aug. 1, 1915.	
						July 1	Aug. 1	Total.	On ship.
	BATTLESHIPS:								
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	94.13	94.79	94.7	94.4
37	*Oklahoma*	New York S. Co.	Reciprocating	2	20.5	96.32	96.55	97.3	97.3
38	*Pennsylvania*	Newport News Co.	Cur. trb. grd. cr.	4	21	75.11	78.57	84.0	80.8
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	46.57	51.15	68.9	65.5
40	*California*	Navy Yard, N. Y.	Electric	4	21	...	...	...	...
41	*Mississippi*	Newport News Co.	Cur. trb. grd. cr.	4	21	4.61	.7.96	25.7	9.9
42	*Idaho*	New York S. Co.	Pars. trb. grd. cr.	4	21	18.82	23.64	35.5	22.1
	DESTROYERS:								
53	*Winslow*	Wm. Cramp & Sons	Cramp trb. & rec.	2	29	96.92	97.95	97.6	97.6
55	*Cushing*	Fore River S. Co.	Curtis trb. grd. cr.	2	29	97.38	97.38	98.3	98.3
56	*Ericsson*	New York S. Co.	Pars. trb. & rec.	2	29	98.00	99.33	100.0	100.0
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	66.82	79.36	79.3	76.7
58	*Conyngham*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	79.41	81.48	82.2	80.9
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	77.18	78.99	76.8	75.4
61	*Jacob Jones*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	84.08	88.12	83.7	83.7
62	*Wainwright*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	84.00	87.93	83.5	83.5
63	...	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	40.57	51.98	36.7	31.4
64	...	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	40.93	44.50	34.2	28.1
65	...	Bath Iron Works	Pars. trb. grd. cr.	2	30	26.21	31.58	32.7	27.3
66	...	Bath Iron Works	Pars. trb. grd. cr.	2	30	26.21	30.07	31.1	25.5
67	...	Wm. Cramps & Sons	Pars. trb. grd. cr.	2	29.5	22.70	24.71	15.4	7.1
68	...	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	6.15	8.8	4.8	4.4
	FUEL SHIPS:								
13	*Kanawha*	Navy Yard, Mare Isl'd	Reciprocating	2	14	99.95	99.95	...	...
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	65.47	69.65	95.1	94.7
15	*Cuyama*	Navy Yard, Mare Isl'd	Reciprocating	2	14	...	...	...	...
	SUBMARINES:								
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	93.00	93.00	88.6	88.4
40	*L-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.4	98.4
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.4	98.4
42	*L-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96.89	96.89	93.3	93.3
43	*L-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.99	95.99	92.6	92.6
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	12.00	12.7	74.4	69.8
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	8.97	10.61	64.1	60.9
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	8.97	10.62	61.9	57.9
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	91.66	92.79	83.6	81.1
48	*L-8*	Navy Yard, Portsmouth	Diesel-Sulzer	2	14	4.84	3.42	43.6	38.9
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	89.92	89.63	76.6	72.2
50	*L-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	88.52	89.29	73.2	67.8
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	78.95	87.05	67.8	61.2
52	*Schley*	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	...	...	...	...
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	10.6	11.2	...
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	10.6	11.2	...
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	...	10.6	11.2	...
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	4.0	29.7	27.1
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	...	4.0	29.1	26.8
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	—	4.0	27.7	25.3
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	—	4.0	28.0	25.7
	SUBMARINE TENDER:								
2	*Bushnell*	Seattle Con. & D. D. Co.	Pars. trb. gearing	1	14	95.30	98.0	92.7	89.7
	DESTROYER TENDER:								
2	*Melville*	New York S. Co.	Pars. trb. gearing	1	15	95.16	97.63	99.2	99.2
	Transport	Navy Yard, Phila.	Reciprocating	2	14	5.51	9.33	21.0	13.5
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	6.85	9.83	19.9	14.2
	TUGS:								
17	...	Navy Yard, Charleston	Recip. oil fuel	1	11	—	—	...	—
18	*Pocahontas*	Navy Yard, Norfolk	Recip. oil fuel	1	11	—	—	...	—
	Ferry Launch	Navy Yard, Charleston	Reciprocating	1	...	—	—	...	—

* *Wadsworth* delivered July 23, 1915.

COMPLETED SHIPS AND RELATIVE STRENGTH OF THE CONTENDING NAVIES.

Type of vessel.	Completed ships of the contending navies, August 1st, 1914, including warships of 1,500 tons or more, and torpedo craft of over 50 tons.										Relative strength of the Triple Entente and Dual Alliance.			
	Great Britain.		France.		Russia.		Germany.		Austria.		Triple Entente.		Dual Alliance.	
	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.	No.	Tons.
Dreadnoughts........	31	661,650	4	92,368	...	....	16	351,519	3	60,030	35	754,018	19	411,549
Pre-dreadnoughts....	40	589,385	18	262,675	7	98,750	20	242,800	6	74,613	66	950,810	26	317,413
Coast-defense ships..	.....	....	1	8,800	2	10,380	2	8,168	6	41,700	3	19,180	8	49,868
Armored cruisers.....	34	406,800	20	201,724	6	63,500	9	94,245	2	13,380	60	672,024	11	107,625
Cruisers..................	74	382,815	9	46,095	9	52,845	41	150,747	5	13,815	92	481,755	46	164,562
Destroyers	167	125,850	84	35,812	91	36,748	130	67,094	18	9,450	342	198,410	148	76,544
Torpedo boats.........	49	11,488	135	13,426	14	2,132	.....	....	39	6,852	198	27,046	39	6,852
Submarines............	75	30,362	64	27,940	30	6,506	21	14,140	6	1,686	169	64,408	27	15,826
Total tons comp't'd..	.....	2,208,350	.....	688,840	...	270,861	.....	928,713	...	221,526	.....	3,168,051	.....	1,150,239

—"Scientific American."

OBITUARY.

CHIEF ENGINEER BENJAMIN F. ISHERWOOD.

By Rear Admiral George W. Baird, U. S. N., Retired.

Benjamin F. Isherwood was born in New York City on October 6, 1822, and died in that city on June 19, 1915. He was the son of a practicing physician who died at an early age. His mother afterwards married a Civil Engineer.

Mr. Isherwood was graduated at the Albany Academy, in the class with Roscoe Conkling, who was his life-long friend.

He began his career as engineer on the Albany and Schenectady Railroad, with David Matthews, Master Mechanic, after which he was employed in the office of his stepfather on the construction of the Croton Aqueduct for New York City, and later on the Erie Railroad. Mr. Isherwood designed the light-house on Sankaty Head, Nantucket, when a mere youth. He was sent to France by the Treasury Department to superintend the construction of lenses of his own design and to make an examination and report on the light-houses of France.

When the Engineer Corps of the Navy was created, in 1844, Mr. Isherwood, in order to qualify, worked with machine tools in the Novelty Works at New York. He entered the Navy as a First Assistant Engineer and was promoted to be Chief Engineer in 1848.

During the war with Mexico he saw service on board the *Princeton*, the first screw-propeller ship of the Navy, and on board the *Spitfire*, and was in every action in which the ships of the Navy were engaged in the Gulf.

He served at the Pensacola Navy Yard in 1844–5; on board the *General Taylor* in 1047–8; in the Navy Department in 1852–3, when he designed the first feathering paddle-

wheels—for the *Water Witch*—ever used in the Navy. He was Chief Engineer of the *San Jacinto* from 1854 to 1858, on the coast of Africa and in the East Indies. He was on special duty in Washington in 1859–60, during which time he made complete designs for a class of gunboats for the Russian Government.

In 1861 he was appointed Engineer-in-Chief of the U. S. Navy, and when the Bureau of Steam Engineering was created in 1862, he became its first Chief.

When the Civil War began, in 1861, sailing ships were still being built for the Navy. The Steam Navy consisted of six frigates of small power, six sloops of war, nine gunboats, two despatch boats and five side-wheel vessels of low power. Designs for ships and machinery were made in the Navy Department as rapidly as possible and mercantile vessels which appeared at all suitable were purchased and armed. At the termination of the war the Navy had about 600 vessels of all classes in commission. Under these trying conditions, Mr. Isherwood worked from 16 to 20 hours a day. He not only gave the leading dimensions for the engines, but scrutinized every detail drawing as it progressed, making such frequent changes that the draftsmen (Assistant Engineers of the Navy) were rarely able to claim even a detail as their own design. The Engineers of the Navy were at that time selected from the engine builders, for there were neither schools for mechanical engineers nor any but the most elementary literature on the subject. In the matter of designing, as in scientific research, Mr. Isherwood led; though he selected the flower of the corps for his assistants, he led them all. He conducted numerous scientific and original investigations to enable him to solve the previously unsolved problems which confronted the designer. He was the first who had the courage to attack and disprove the vicious theory of Marriott and Gay-Lussac, and though this brought down on him invectives from the leading engineers of Europe and America, he lived to see them turned into encomiums.

It was from the printed results of his researches, which were

soon confirmed by the admirable works of Drs. Tyndal and Mayer, that the advantages of high-pressure, multiple-expansion engines were made apparent and the loss by cylinder condensation diminished. In short, it was from the results of these researches that ships could be built to cross the Atlantic in less than a week.

Mr. Isherwood made a mark upon the times which few have equalled in any branch of science. The Confederate cruiser *Alabama*, a nine-knot boat, had so far depleted the commerce of the North that there was but little of it left. Though we owned and operated many vessels capable of making 12 knots, and some 15 knots, none of them could catch that evasive cruiser. The demand for greater speed was loud, long and potent. Mr. Isherwood then designed machinery for an 18-knot class of vessel. Three vessels of this class were built and all of them attained the designed speed, although some of the best European engineers had declared it impossible. As the vessels had wooden hulls, a speed of 18 knots would, in fact, have been impossible had not the weight of machinery, battery, etc., been so distributed in the vessel as to prevent any straining of the hull.

"M. V. Dwelshanvers-Derry, the eminent Belgian engineer and man of science," says Dr. Thurston, "now perhaps the leading authority in his department," refers to Commodore Isherwood as "*l' eminent ingenieur en chef de la marine des Etats Unis, le plus fecond des experimenteurs de ces quarante dernieres annees.*"

The two volumes of Isherwood's Experimental Researches have been translated into six European languages.

He was the first to show the proper way to test an engine and determine where all the losses lay.

He possessed not only a wonderfully retentive memory, but also the faculty of "statement" which enabled him to make complicated questions clear to the untutored mind.

Mr. Isherwood was the personification of dignity, politeness and affability.

He married Miss Anna Hausine Münster, to whom six children were born.

In 1873 he purchased a house in 36th Street, New York City, and, taking advantage of the exile of the French communists at that time, employed the best artists among them to decorate it. The walls are artistically painted in oil colors and in gold, and all the furniture and fittings match, making it perhaps as artistic, if not as elegant, a residence as may be found in the city.

Chief Engineer Isherwood will ever be revered by all who knew, honored and loved him for himself and the great work he accomplished.

The frontispiece for this number of the JOURNAL is from a photograph which was taken when Mr. Isherwood was seventy years of age.

ALBERT LLOYD HOPKINS.

Albert Lloyd Hopkins was born September 7, 1871, at Glens Falls, New York, the son of Stephen DeForest and Elizabeth G. Hopkins. His early life was passed and his early education was received in that city and in Troy, New York.

In 1888 he entered the famous Rensselaer Polytechnic Institute in the course of civil engineering. In 1892 he was graduated with high honors, and in later years the Director of the Institute said of him: "About ten per cent. of those who apply are admitted to the Institute, about twenty per cent. of those admitted are graduated, and among these, once in a while, we find a Hopkins."

After a few months' work in the summer of 1892 in an architect's office in Chicago, he was appointed to a position in the Bureau of Construction and Repair, Navy Department, Washington, D. C., which position he held for about eighteen months, coming to Newport News in February, 1894, as a member of the staff of Naval Constructor J. J. Woodward, U. S. N., the first Superintending Constructor assigned to duty

at the works of the Newport News Shipbuilding and Dry Dock Company.

In the summer of 1897 Mr. Hopkins was transferred from Naval Constructor Woodward's office and assigned to the Graduate School of Naval Architecture at the U. S. Naval Academy, Annapolis, Maryland, at which place he was an instructor and lecturer on naval architecture and ship construction.

At the outbreak of the Spanish War, in April, 1898, the officers and students of this school joined the Fleet, and Mr. Hopkins was assigned to the Naval Station at Key West, Florida, the nearest station to the blockading fleet operating in Cuban waters. While at Key West Mr. Hopkins was in charge of all the construction and repair work done for the Fleet at that station. He was also active in improving the efficiency of the plant, installing much new equipment and rendering it capable of serving as a repair station for naval purposes.

It was while engaged in the work at Key West that Mr. Hopkins received from the late W. A. Post, then General Superintendent of the shipyard, an offer to return to Newport News and enter the service of the company. Mr. Hopkins accepted and in August, 1898, returned to Newport News as the personal assistant of Mr. Post.

The able constructive work done by Mr. Hopkins in the years following will always be remembered by those associated with him. Gifted with a rarely keen mind, he was quick to grasp the essentials of any situation. His education and training enabled him to choose unerringly the right course to pursue, and his strong will and personality enabled him to carry to its logical conclusion the course so chosen. United with these strong characteristics were an unfailing courtesy and a consideration for others which endeared him to the hearts of all his associates.

When Mr. Post was made General Manager of the company, in 1905, Mr. Hopkins was appointed Assistant General Manager, and in 1911, when Mr. Post succeeded to the Presidency,

on account of the death of Mr. C. B. Orcutt, Mr. Hopkins was made Manager.

Upon the death of Mr. Post, in February, 1912, Mr. Hopkins was elected Vice President and became the chief executive officer of the company, with headquarters in New York.

In this new field Mr. Hopkins fully sustained the high position accorded to his predecessors, and it was quickly and widely recognized that an able executive was directing the affairs of the Shipbuilding Company. In his own profession, and also among the men his profession called him to meet—bankers, lawyers and men of large business affairs—his influence was deep and ever increasing.

His election to the Presidency of the company, in March, 1914, was recognized as a well earned tribute to his ability and his devotion to the interests committed to his care.

Loyal to his friends and associates, loyal to those who trusted their interests to him, loyal always to his own high ideals, well and truly does he deserve the tribute expressed in the following telegram sent by Mr. Huntington to the company.

"Mrs. Huntington and I are distressed beyond expression at the death of Mr. Hopkins. We believe the company and all the employees share our sorrow and will mourn the loss of a splendid officer and a noble man."

To those who knew him best, to his friends and associates of many years, the tidings of the disaster brought a sure message; well they knew that the gentle, kindly courtesy, the ever present, instinctive disposition to serve others before considering himself would unfailingly include him among those who died that others might live.

Mr. Hopkins was a member of the American Society of Civil Engineers, the American Society of Naval Engineers, the Society of Naval Architects and Marine Engineers, and the American Academy of Political and Social Science.

In June, 1906, Mr. Hopkins was married to Miss May Davies, of Chase City, Virginia, who, with their daughter, May Davies Hopkins, survives him.

Mr. Hopkins is also survived by his mother, of Glens Falls,

New York, and his brother, Charles E. Hopkins, of Hudson, New York. Mr. Hopkins' father, Stephen DeForest Hopkins, died at his home in Glens Falls May 1, 1915, only a few hours after his son sailed on the ill-fated voyage.

CHIEF ENGINEER ROBERT CRAWFORD, U. S. N.

Chief Engineer Robert Crawford died at Lansdowne, Pa., on the 8th of April, 1915, of ticdouloureux, from which he had been a sufferer for two years.

He was born in Scotland in 1842, and came to the United States when four years of age.

He was graduated with honor at the High School in Philadelphia, and became a draughtsman in the Baldwin Locomotive Works, where he remained until appointed a Third Assistant Engineer in the Navy, his appointment dating 23d June, 1863.

He served through the remainder of the Civil War, on board the *Chippewa*, *Monitor*, *Pawnee* and *Periwinkle*, in the North Atlantic Squadron, taking part in the battles of Wilmington, Fort Fisher, etc. He was engineer of the famous iron clad *Stonewall*, after she was turned over to the United States at Havana, and he brought her to Washington. He served on board the *Madawaska*, *Swatara*, *Monongahela*, *Kearsarge* and the *Alert*, and was Instructor in Experimental Philosophy, and also in Steam Engineering, at the Naval Academy, for periods aggregating about six years.

He was retired in 1902, unable to longer endure sea duty. He organized the Williamson School of Manual Training, and was professor there for eleven years, after his retirement. He organized the Manual Training and Reform School at Guanapay, Cuba, and, after it was running smoothly, he became Professor in the Academy at Santiago, de las Vegas.

During the war with the Kingdom of Spain Mr. Crawford was detailed as Inspector of Machinery at the Cramps' Ship Yard, at Philadelphia, where he remained until 1912, when he gave up all active employment.

Mr. Crawford possessed a genial personality, was a good after-dinner speaker, a kind father and an indulgent husband. He leaves a widow, two sons and one daughter. He was an active member of St. John's P. E. Church, at Lansdowne. He was, for years, Chief Burgess of Lansdowne; was a member of the Union Athletic Association, a member of the Loyal Legion, the American Society of Naval Engineers and of the Masonic Fraternity. He was active in all patriotic movements.

In politics he was a Republican, but never took a very active part in its activities. He had a never-ending supply of humorous stories, and he was held in high esteem by his shipmates.

Mr. Crawrord was ever a painstaking and conscientious man, faithful to his trust, and inflexible in his fidelity.—G. W. B.

PUBLICATIONS RECEIVED.

FLEETS OF THE WORLD.—Contains a short glossary of Naval terms; comparative tables of the large-caliber guns of the great Fleets of the world; a list of ships lost in the war from August 5, 1914, to April 15, 1915. Over 100 illustrations. Published by J. B. LIPPINCOTT COMPANY, Philadelphia. Cloth, $2.50.

NATIONAL BULLETIN NO. 20.—An index for National Bulletins Nos. 1–20, offering reliable pipe information that is readily accessible. NATIONAL TUBE COMPANY, Pittsburgh, Pa.

ASSOCIATION NOTES.

THE FOLLOWING MEMBERS AND ASSOCIATES have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Baxter, Thomas, Lieutenant, U. S. Navy.
Blasdell, Francis G., Ensign, U. S. Navy, Ret.
Border, Lee S., Assistant Naval Constructor, U. S. Navy.
Burrough, Edmund W., Ensign, U. S. Navy.
Butler, William J., Ensign, U. S. Navy.
Davis, Arthur C., Ensign, U. S. Navy.
Deets, Samuel R., Ensign, U. S. Navy.
Grady, Ronan C., Lieutenant, U. S. Navy.
Haines, Preston B., Lieutenant, U. S. Navy.
Hird, Harry B., Lieutenant, U. S. Navy.
Kemp, Dennis E., Lieutenant, U. S. Navy.
Kessing, Oliver O., Ensign, U. S. Navy.
Kimberly, Victor A., Lieutenant Commander, U. S. Navy.

Laizure, Dallas C., Lieutenant, U. S. Navy.
Lamont, George P., Ensign, U. S. Navy.
Lanphier, Alfred Y., Ensign, U. S. Navy.
Leigh, Richard H., Commander, U. S. Navy.
McWhorter, Ernest D., Lieutenant, U. S. Navy.
Mallison, William T., Lieutenant, U. S. Navy.
Mann, Joseph R., Jr., Lieutenant, U. S. Navy.
Molten, Robert P., Lieutenant, U. S. Navy.
Nimitz, Otto, Ensign, U. S. Navy.
Oates, Eugene T., Ensign, U. S. Navy.
O'Brien, William H., Jr., Lieutenant, U. S. Navy.
Oman, Joseph W., Captain, U. S. Navy.
Parsons, Henry E., Lieutenant, U. S. Navy.
Quinlan, Earl H., Ensign, U. S. Navy.
Wagner, Frank D., Ensign, U. S. Navy.
Walker, Myron J., Ensign, U. S. Navy.
Welden, Fred, Lieutenant, U. S. Navy.
Withers, Noble, Ensign, U. S. Navy.
Wright, Carrol Q., Jr., Ensign, U. S. Navy.

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JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XXVII. NOVEMBER, 1915. No. 4.

THE MYSTERY OF THE SCREW PROPELLER.

BY CAPTAIN C. W. DYSON, U. S. NAVY, MEMBER.

For many years after the adoption of the screw propeller for the propulsion of ships the seeming vagaries in its performances in actual service of propulsion have cast a mystery over it and over the laws governing its action.

The greater part of this mystery is, however, not due to the propeller, but can be directly placed on the carelessness with which trials of ships have been run and data collected, while the greater part of the remainder can be attributed to the effect of variations in hull form in interfering with the proper flow of water to the propeller, thus seriously decreasing the propulsive efficiency; in producing more or less wake and so adding to the propulsive efficiency; in incorrect estimates of effective horsepowers required for given speeds, basing these powers on estimates of frictional and residual resistances with an entire neglect of the malign influence of the appendages such as struts, bilge keels, etc.

The small remaining portion of the mystery can be ascribed

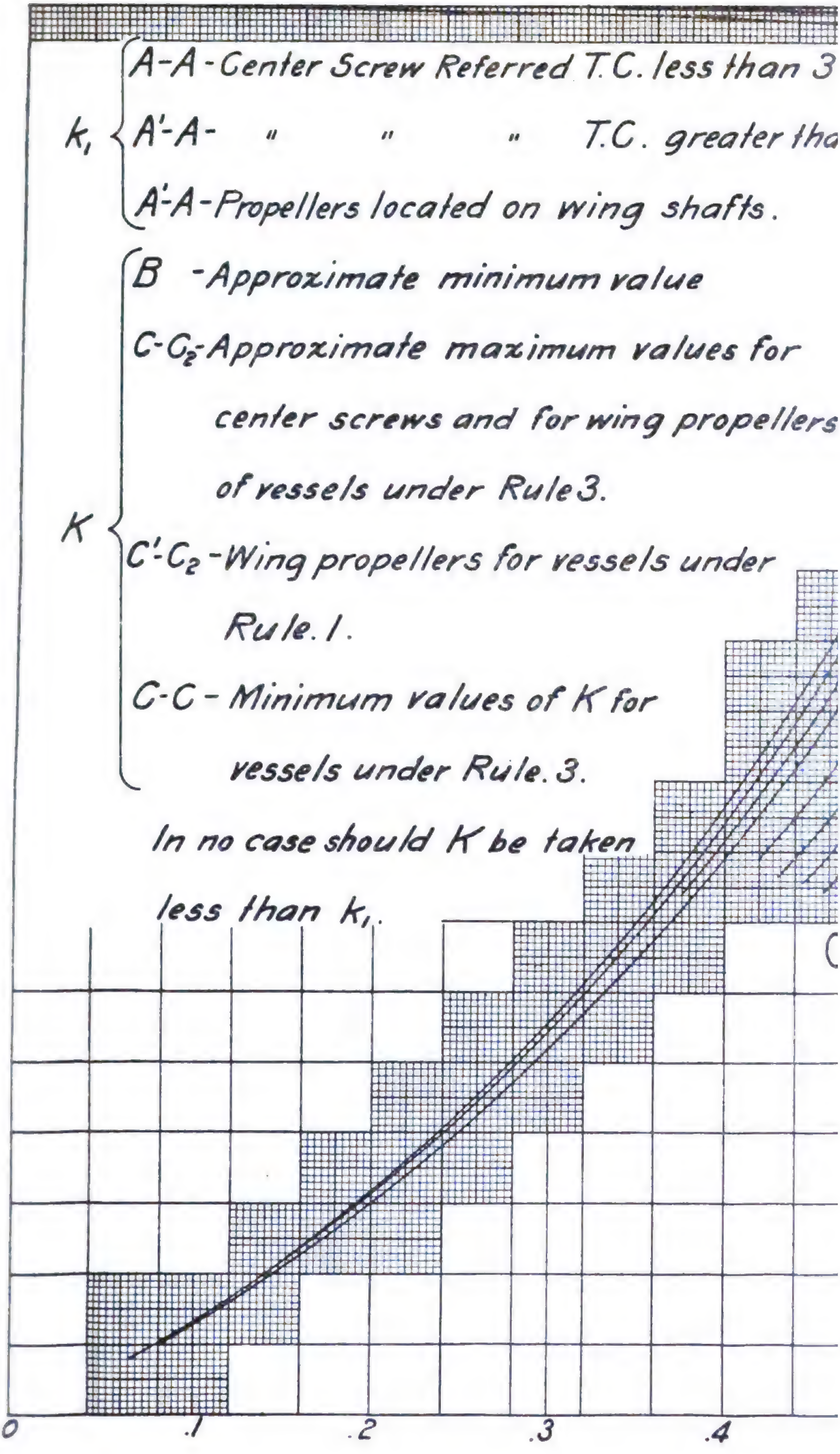
to the propeller itself, and is due to the myriad variations in blade forms and sections which have been used, these appearing to depend upon the taste of the designer, and upon the lack of a consistent basis of comparison for the analysis of propeller performances.

As the years rolled by they brought in their wake the model tank by which a more nearly accurate value of the effective horsepower can be obtained; more accurate instruments for the measurement of indicated and shaft horsepowers; better mechanical construction of propelling engines by which frictional losses have been greatly reduced and brought to a more nearly constant and even value; more care in running trials over the measured mile combined with a better knowledge of the effect of shallow water on the course, both tending towards the production of better speed data; and these improvements have resulted in the production of data from which curves of analysis may be drawn, rendering it possible to take any propeller which may be proposed for any given vessel, and to estimate a curve of performance for any condition of loading with more than a fair degree of accuracy.

K—CORRECTIVE FACTOR FOR INDICATED AND SHAFT HORSEPOWERS.

When a propeller works at the stern of a vessel it operates in a body of water which partakes, to a more or less degree, of the forward movement of the vessel itself. When the propeller is so located that the suction column of water entering the propeller disc enters normal to the disc and with very little disturbance, and when the propeller blade tips are well immersed and pass the hull at a good distance from it, the wake will increase the effective thrust of the propeller, and therefore the effective horsepower delivered, for any given indicated or shaft horsepower which may be delivered by the propelling engine. This gain is known as the *wake gain*.

Should the propeller be so located in relation to the hull



k_1
A-A - Center Screw Referred T.C. less than 3
A'-A - " " " T.C. greater tha
A'-A - Propellers located on wing shafts.
K
B - Approximate minimum value
C-C_2 - Approximate maximum values for center screws and for wing propellers of vessels under Rule 3.
C'-C_2 - Wing propellers for vessels under Rule. 1.
C-C - Minimum values of K for vessels under Rule. 3.
In no case should K be taken less than k_1.
0
.1
.2
.3
.4

that the suction column is no more a cylinder but becomes a frustrum of a cone, the propeller disc being the upper bounding plane, or should the propeller blades be insufficiently immersed or pass unduly close to the hull, or should combinations of the above conditions exist, the effective thrust, and consequently the effective horsepower, per revolution for any indicated or shaft horsepower delivered by the propelling engine, will be reduced. This loss is commonly called the *thrust deduction.*

In cases where the thrust deduction exceeds the wake gain, and such cases are the usual ones, the result is a net loss in propulsive efficiency and an increase in revolutions for the given speed.

Where the wake gain exceeds the thrust deduction, and such cases are rare, the opposite effects are produced.

The factor K represents unity plus the net loss or minus the net gain, as the case may be.

CONTROL OF THE VALUE OF K.

As the value of K is fixed by the lines of the hull and the relative positions of propeller and hull, there exists a certain amount of freedom in fixing the loss due to this relative position. By practical considerations which are forced, the propeller cannot be further removed aft from the fullness of the lines than a certain amount for any hull, these considerations being governed by the necessity for the shaft and propeller supports, and this maximum distance, fore and aft, fixes the minimum value of K for any hull. This value of K will then be increased as the tip clearance between hull and propeller decreases below a certain amount, this amount depending upon the slip block coefficient and upon the height of the horizontal line of least tip clearance, usually the height of the hub center above the base line of the vessel. The amount of immersion of the upper tips of the blades below the water surface appears to have some influence, but very slight as compared with

tip clearance for vessels acting on the surface. In the case of submarines, however, where the hydraulic head becomes very much increased by diving, the depth of immersion increases in importance and undoubtedly adds to the *propulsive efficiency* of the propeller.

The full value of K is made up of two quantities, which may be called k_1 and k_2. k_1 is the natural loss due to the hull itself, affecting the density of the water in the column flowing to the propeller, the propeller being well located and with good tip clearance. k_1, on Fig. 1, is represented by the curve marked A. k_1 requires increased power and increased revolutions to deliver any given effective horsepower. k_2, on the other hand, affects the indicated or shaft horsepower only, and is caused by the malign influence of currents of water entering the propeller disc radially and to eddying between the blades, neither adding anything to the thrust but adding directly to the resistance to turning. Therefore, in estimating revolutions for any given speed, k_1, only, is used, as also in computing diameter of the propeller, while for the estimate of final I.H.P. or S.H.P., the full value of $K = k_1 + k_2$ must be used.

On this Figure are shown curves of K, having slip block coefficients as abscissas and mean blade-tip clearances as ordinates. On this same figure are shown cross curves giving the minimum, marked B and maximum C^1 values of K that can be expected with blades of Standard Form and with Standard Hull Form. There are also shown curves of k_1 for propellers located under the quarters of a vessel as well as for those located directly in wake of the stern post.

The term "mean tip clearance" is misleading, as the ordinates are averages of tip clearance and immersion of upper blade tip, both being referred to standard conditions of clearance and immersion and the mean taken by giving each factor its apparent weight in the equation for "mean tip clearance."

This equation is:

Mean tip clearance $= \frac{H + 10\ H^1}{11}$, where

H^1 (Twin, 4, etc., screws) = (Actual horizontal T.C. $\times$ 10) $\div$ Actual height of center of hub above base line.

H^1 (Single or Center Screw) = Fore-and-aft Clearance in feet between center line of vertical blade and fullness of hull.

H (depth of immersion of upper blade tip) = (Actual Immersion in feet $\times$ 14) $\div$ actual height of tip above base line.

RULES FOR APPLYING THE ABOVE.

RULE 1.—*Where blades are of standard form, and in the cases of other than center-line propellers when a vertical line through the propeller hub center penetrates the hull close to or below the load-water plane, apply the curves as they stand.*

RULE 2.—*Where the blades are narrow tipped, as mean tip clearance passes from max. K to min. K, decrease values of K by from .03 at max. to 0 at min. Where broad-tipped or bulbous, increase K by same amounts.*

RULE 3.—*When vertical line through hub center, with other than center-line propellers, pierces hull well above the load-water plane, use curve of K marked C and do not correct slip block coefficient for variation of midship section coefficient.*

THE SCREW PROPELLER.

All screw propellers when working under similar conditions of resistance arrange themselves in one great family in which the position of any particular propeller is fixed by its diameter, its pitch and its projected area ratio, the latter fixing the dimension of the thrusts and the resultant tip speed and propulsive efficiency which may be expected.

As the condition of the resistance changes the values of the thrusts and of the tip-speed change in inverse proportion.

These thrusts, tip speeds and propulsive efficiencies can be shown graphically as curves, the abscissa values of which are

projected area ratios, and the family position of any propeller can be found by using these values to ascertain the indicated or shaft horsepower and the effective horsepower which the propeller will deliver under the conditions of resistance which produce such curves. These conditions will be spoken of as "Chart Conditions."

On Fig. 2 are shown the curves of propulsive coefficients, tip speeds in feet per minute, and indicated thrusts in pounds per square inch of disc area of the propeller for values of projected area ratios from zero to over .6, and may be regarded as accurate from the value .25 to the maximum.

These curves are exact reproductions of the corresponding design curves shown on Chart 5 (Sheet 21) of the author's work on "Screw Propellers" except the portion of the propulsive coefficient curve from a projected area ratio of .56 to the maximum ratios shown; experience in this range has shown conclusively that higher values of propulsive efficiency can be realized in this region than were credited to it in the original curve, and the curve has been corrected and brought up to the values which may be confidently looked for unless peculiarities of hull, abnormal trial conditions, or position of propeller intervene and produce conditions of which no account can be taken in making a forecast of a propeller's performance.

DATA AND FORMULAS FOR FINDING THE CHART CONDITION OF A PROPELLER.

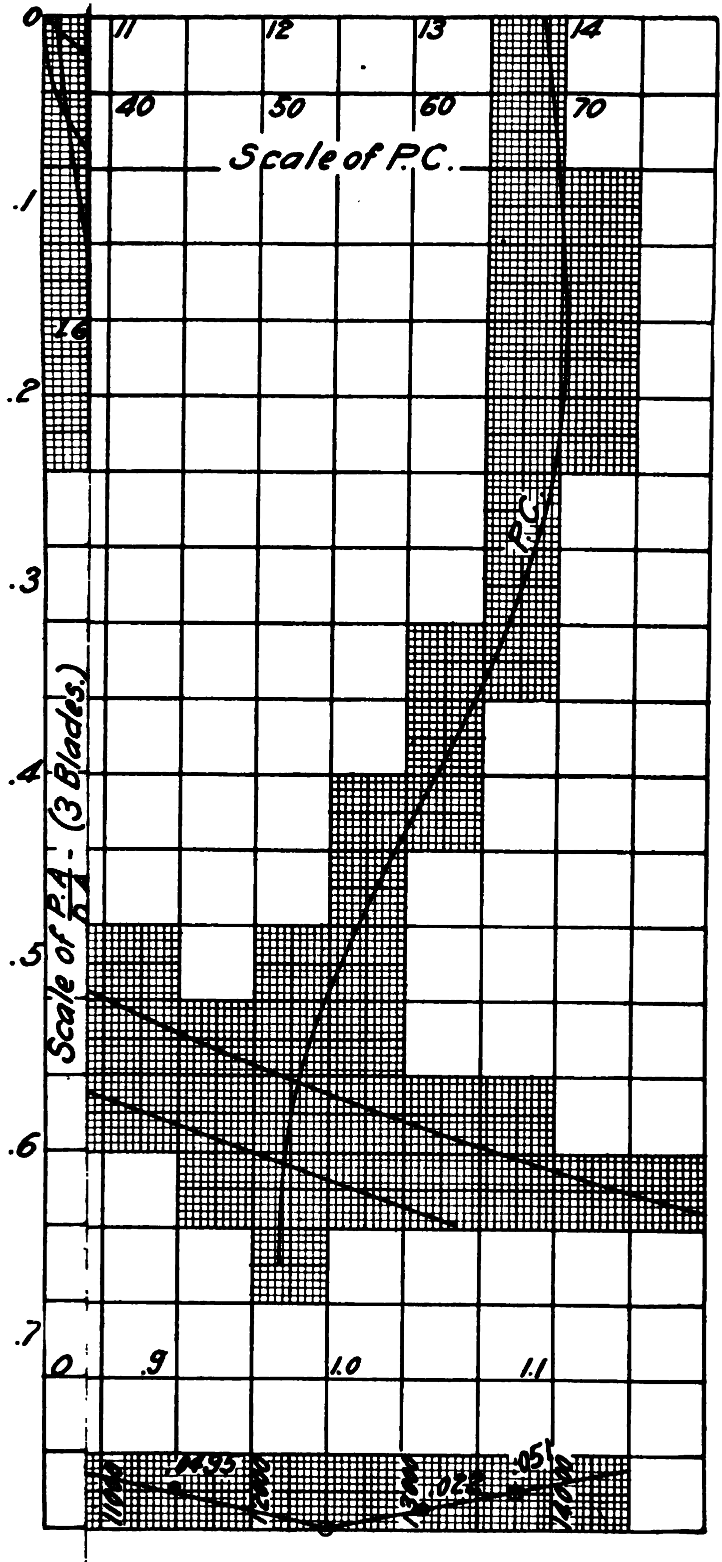
In finding the chart condition of any given propeller the following data must be at hand:

(1) P.A. ÷ D.A. = Actual projected area ratio of a three-bladed, $\frac{3}{4}$ of this ratio for a 4 and $\frac{3}{2}$ of it for a 2-bladed propeller.

(2) P = Pitch of the propeller, in feet.

(3) D = Diameter of the propeller, in feet.

(4) T.S. = Tip speed for P.A. ÷ D.A. (Fig. 2).



ıese points are given.

(5) $R = T.S. \div \pi D$ = Revolutions under chart conditions.
(6) $P \times R$ = Pitch in feet $\times$ revolutions per minute.
(7) $I.T._d$ = Indicated thrust per square inch of disc area or $P.A \div D.A.$ (Fig. 2).
(8) N = Number of propellers working under similar conditions.
(9_a) $I.H.P. = (N \times D^2 \times I.T._d \times P \times R) \div 291.8$ = indicated horsepower (3 blades).
(9_b) $I.H.P. = (N \times D^2 \times I.T._d \times P \times R) \div (291.8 \times .865)$ = indicated horsepower (4 blades).
(9_c) $I.H.P. = (.75 \times N \times D^2 \times I.T._d \times P \times R) \div 291.8$ = indicated horsepower (2 blades) all under chart conditions.
(10) P.C. = Propulsive coefficient for $P.A. \div D.A.$ (Fig. 2).

(In the cases of four and two-bladed propellers, the value of propulsive coefficient given by Fig. 2 for the actual full projected area ratio must be used.)

(11) $E.H.P. = I.H.P. \times P.C.$ = effective horsepower delivered by the propeller under chart conditions.

It will be noted at once that the value of the slip block coefficient of the vessel has no place whatsoever in these formulas, nor does it have any effect of any kind on the work which the propeller can perform when operating under the chart condition.

When the chart condition is departed from and the actual conditions existing in the wake of the hull are considered, the influence of the hull form must be taken into account and the value K is finally brought into the equation for final indicated and shaft horsepower necessary to deliver any value of effective horsepower.

ESTIMATE OF CHART CONDITION OF SPEED, V.

In obtaining this estimate, the slip block coefficient (corrected for midship section coefficient and for position of propeller, as explained in the author's work, "Screw Propellers"),

enters into the data values, as this coefficient affects the value of the effective thrust per square inch of projected area, and the factor $1 - S$, in which S is the apparent slip for the particular slip block coefficient and projected area ratio under consideration, is equal to the propulsive thrust in pounds per square inch of projected area divided by the effective thrust. The data and formulas for obtaining V are:

(12) $P.T._p$ = Propulsive thrust in pounds per square inch projected area for P.A. ÷ D.A. (Chart 5, sheet 21, "Screw Propellers").

(13) $E.T._p$ = Effective thrust in pounds per square inch projected area for P.A. ÷ D.A. and for slip block coefficient (Chart 5, sheet 21, "Screw Propellers").

(14) S = Apparent slip under chart conditions.

(15) $1 - S = P.T._p \div E.T._p$ = per cent. advance of ship under chart conditions.

(16) $V = \left\{ P \times R \times (1 - S) \right\} \div 101.33$ = speed of ship under chart conditions.

TO CHANGE FROM CHART CONDITIONS TO OTHER CONDITIONS OF RESISTANCE.

Suppose a vessel to be so loaded that for the speed V an indicated horsepower I.H.P., or a shaft horsepower, S.H.P., are required to deliver the effective horsepower, E.H.P., necessary to produce this speed, the revolutions of the propeller being R and the tip speed T.S. The propeller is then working under chart conditions of resistance.

Should the speed be reduced by reducing the power of the engines, by increasing the displacement, by fouling of bottom, by condition of wind or sea, etc., or should the opposite effect occur and the speed be increased, the conditions of resistance differ from the chart conditions and the following changes occur in the propeller performance:

Chart Condition of Loading for V, but

Engine power reduced:—

Effective horsepower reduced.

Revolutions reduced.

Tip speed reduced.

Thrusts reduced.

Speed reduced from V to *v*, I.H.P. reduced to I.H.P. and where K is greater than unity, to K $\times$ I.H.P.$_d$.

Load increased for V above Chart Condition.

Chart engine power constant = I.H.P.:—

Effective horsepower = E.H.P.

Revolutions reduced.

Tip speed reduced.

Thrusts increased.

Speed reduced from V to *v*, and where K is greater than unity I.H.P. increases to K $\times$ I.H.P.

Load decreased for V below Chart Conditions.

Chart engine power constant = I.H.P.:—

Effective horsepower = E.H.P.

Revolutions increased.

Tip speed increased.

Thrusts decreased.

Speed increased from V to *v*, and where K is greater than unity I.H.P. becomes K $\times$ I.H.P.

It will be noted in these changes of conditions that so long as the I.H.P. remains constant the E.H.P. remains constant also, no matter how the actual speed of the ship may change. This statement is borne out by comparison of trial results of several vessels where the vessels were of sufficiently fine after body and propellers were so well placed as to practically assure that K equalled unity. In these cases the agreement between

the actual indicated, shaft, and effective horsepowers, and those of the chart condition of the propeller were so close as to justify the following statement:

RULE 4.—*Should a screw propeller working in the wake of a vessel deliver a certain effective horsepower with a certain indicated or shaft horsepower under any given condition of resistance, it will deliver the same effective with the same indicated or shaft horsepower under any other condition of resistance so long as it is operating in the wake of the same hull, and so long as the effective thrusts are below the point of cavitation.*

This law renders it possible, where a vessel has been tried up to and beyond the speed for which the effective horsepower is equal to the Chart Condition E.H.P. of the propeller used, to obtain the value of K at once as—

The actual indicated horsepower required to deliver E.H.P. = K × I.H.P. (Chart Condition), from which K at once results

$$K = \frac{\text{actual indicated horsepower}}{\text{I.H.P. (Chart Condition)}} = \frac{\text{Actual S.H.P.}}{.92 \text{ I.H.P. (Chart cond.)}}$$

DETERMINATION OF POWER FACTOR Z.

In arriving at a satisfactory series of values of condition factors to use in estimating power due to changes in conditions of resistance from the Chart Conditions, many different forms of equations were tried, using the measured-mile trial data of very long vessels, tried in deep water, and where the trials were conducted in such a manner as to create confidence in the trial results tabulated. Thus all the trials that have been used have had at least three runs for each plotted point of the speed-revolution and speed-power curves while the highest point of the curve has been obtained by five runs. In obtaining the mean of each set of runs, the following method of averaging was used:—

For a three-run point:

Run No. 1.	North.	1 × Power.	1 × Revolutions.
Run No. 2.	South.	2 × Power.	2 × Revolutions.
Run No. 3.	North.	1 × Power.	1 × Revolutions.
Mean, ...		$\frac{\text{Sum}}{4}$	$\frac{\text{Sum}}{4}$

For a five-run point:

Run No. 1.	North.	1 × Power.	1 × Revolutions.
Run No. 2.	South.	2 × Power.	2 × Revolutions.
Run No. 3.	North.	2 × Power.	2 × Revolutions.
Run No. 4.	South.	2 × Power.	2 × Revolutions.
Run No. 5.	North.	1 × Power.	1 × Revolutions.
Mean, ...		$\frac{\text{Sum}}{8}$	$\frac{\text{Sum}}{8}$

The final form of the estimating power equation became:

$$\text{I.H.P.}_d = \text{I.H.P.}\left(\frac{v}{V}\right)^w,$$

where v is any speed and I.H.P._d the indicated horsepower for this speed. When K for the vessel under consideration exceeds unity, the actual indicated horsepower for $v = \text{K} \times \text{I.H.P.}_d$.

Designating the effective horsepower necessary to obtain the speed v, by *e.h.p.*, and taking values of v for several trial vessels for the load ratios $\frac{e.h.p.}{\text{E.H.P.}} = .025, .05, .075, .1, .2, .3, .4$, etc., up to 1.1, and solving the equation

$$w = (\log \text{I.H.P.} - \log \text{I.H.P.}_d) \div (\log \text{V} - \log v),$$

I.H.P._d being the actual indicated horsepower for v, where $\text{K} = 1$, and being equal to the actual indicated horsepower for v divided by K where $\text{K} > 1$, a series of curves are obtained for the different load ratios given.

Taking $z = w(\log \text{V} - \log v)$, it was found that for each curve of w, depending on the load ratio $\frac{e.h.p.}{\text{E.H.P.}}$, z had practically a constant value.

These values of z are shown as a curve on Fig. 2, having the ratios $\frac{e.h.p.}{\text{E.H.P.}}$ as abscissas, and they vary in value from ∞ when $\frac{e.h.p.}{\text{E.H.P.}} = 0$, to zero when $\frac{e.h.p.}{\text{E.H.P.}} = 1$, and then again increase in value as $\frac{e.h.p.}{\text{E.H.P.}}$ becomes greater than unity. In this latter case, however, z changes sign. The power equation therefore can be simplified to the forms:

(*a*) . . $\frac{e.h.p.}{\text{E.H.P.}} < 1$, $\log \text{I.H.P.}_d = \log \text{I.H.P.} - z$,
z being positive.

(*b*) . . $\frac{e.h.p.}{\text{E.H.P.}} > 1$, $\log \text{I.H.P.}_d = \log \text{I.H.P.} - z$,
z being negative.

and the final indicated horsepower for any speed v becomes $= \text{K} \times \text{I.H.P.}_d$, and is absolutely independent of speed but depends entirely upon the load ratio $\frac{e.h.p.}{\text{E.H.P.}}$ for its value.

While w is of no further use in obtaining the estimate of power, z having usurped its place, it is of interest to obtain its values for different values of $\frac{v}{\text{V}}$ for varying ratios of $\frac{e.h.p.}{\text{E.H.P.}}$, and then to plot the values corresponding to each value of $\frac{e.h.p.}{\text{E.H.P.}}$ as a curve, using the corresponding values of $\frac{v}{\text{V}}$ as abscissas.

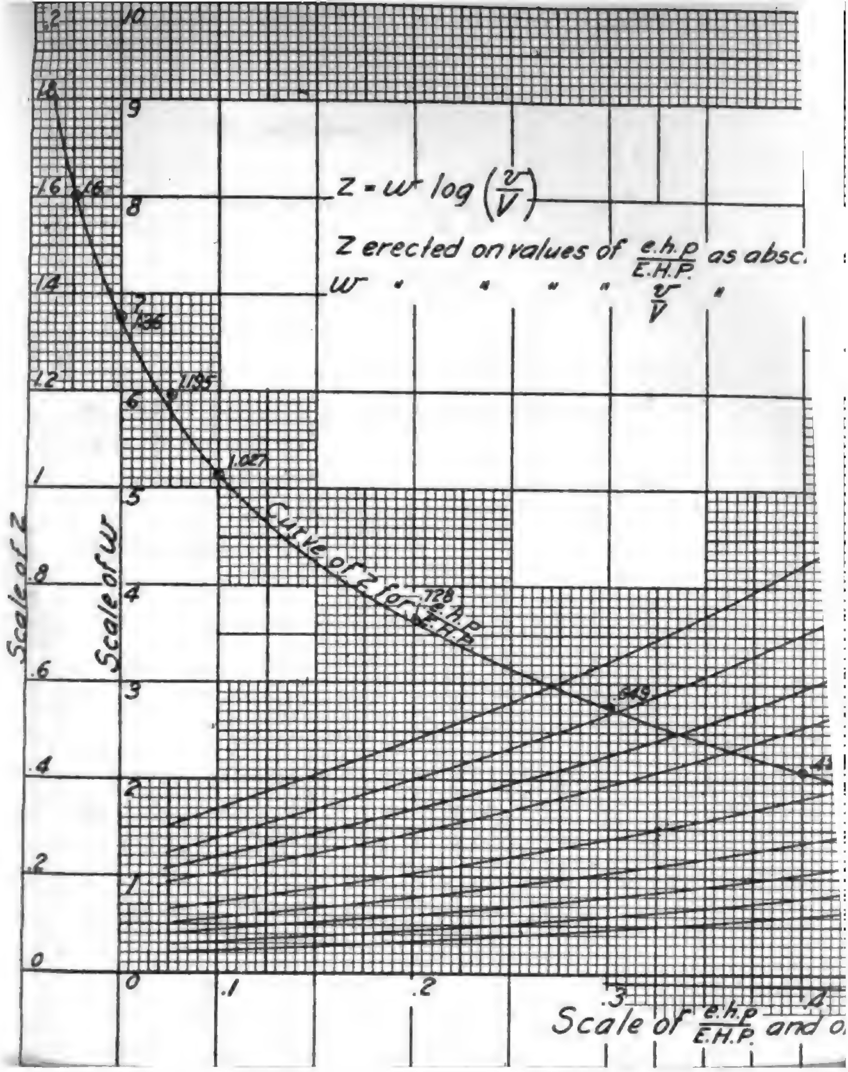
There results from such plotting, two complete series of rectangular hyperbolas, Fig. 3, one for conditions where

$$\frac{e.h.p.}{\text{E.H.P.}} < 1 \text{ and one where } \frac{e.h.p.}{\text{E.H.P.}} > 1,$$

the rectangular asymptotes intersecting each other at

$$\frac{v}{\text{V}} = 1.$$

Thus, in the first quadrant and the third quadrant, opposite branches of the same hyperbolas, there will be



$Z = w \log\left(\frac{v}{V}\right)$
Z erected on values of $\frac{e.h.p}{E.H.P.}$ as absc
w " " " " $\frac{v}{V}$ "
Curve of Z for $\frac{e.h.p}{E.H.P.}$
Scale of Z
Scale of W
Scale of $\frac{e.h.p}{E.H.P.}$ and o
1.6
1.36
1.195
1.027
.728
.649

$$(\text{1st quad.})\ \frac{e.h.p.}{\text{E.H.P.}} < 1, \frac{v}{\text{V}} < 1.$$

$$(\text{3d quad.})\ \frac{e.h.p.}{\text{E.H.P.}} < 1, \frac{v}{\text{V}} > 1.$$

The equations, when conditions fall within either of these quadrants, take these forms (1st quad.) $\log \text{I.H.P.}_d = \text{I.H.P.} + w(\log v - \log \text{V})$,

(3d quad.) $\log \text{I.H.P.}_d = \log \text{I.H.P} - w(\log v - \log \text{V})$.

In the 2d and 4th quadrants, which may be called the overload quadrants,

$$(\text{2d quad.})\ \frac{e.h.p.}{\text{E.H.P.}} > 1, \frac{v}{\text{V}} > 1.$$

$$(\text{4th quad.})\ \frac{e.h.p.}{\text{E.H.P.}} > 1, \frac{v}{\text{V}} < 1.$$

The equations then become for the

(2d quad.). $\log \text{I.H.P.}_d = \log \text{I.H.P.} + w(\log v - \log \text{V})$.

(4th quad.). $\log \text{I.H.P.}_d = \log \text{I.H.P.} - w(\log w - \log \text{V})$.

ESTIMATION OF REVOLUTIONS FOR CONDITIONS OTHER THAN CHART CONDITIONS.

In estimating change of revolutions caused by change of conditions, the following formula, derived from actual trial results, can be used and confidence can be placed in its accuracy:—

$$\text{R}_d = \text{R}k_1^3 \times \left(\frac{v}{\text{V}}\right)^x$$

where R = Chart Revolutions.

k_1 = Minimum value of K for slip B.C.

x = Coefficient for $\frac{v}{\text{V}}$ and $\left(\frac{e.h.p.}{\text{E.H.P.}}\right)$, Fig. 4.

The equation for diameter under these conditions becomes $\text{D} = \sqrt{(291.8 \times \text{I.H.P.}) \div (\text{I.T.}_\text{D} \times \text{P} \times \text{R})}$, where I.H.P.

= the Chart Condition of the propeller for power. Calling this diameter as affected by k_1, D_{k_1},

$$D_{k_1} = D$$
$$P_{k_1} = P$$
$$R_{k_1} = k_1^3 R$$

where D, P and R would be those necessary when $k_1 = 1$.

The equation for revolutions justifies the following statement:

RULE 5.—*If a screw propeller working in the wake of a vessel will deliver a certain effective horsepower with a certain indicated or shaft horsepower at a certain number of revolutions at a certain speed of vessel under any given condition of resistance, it will deliver the same effective horsepower, with the same indicated or shaft horsepower, while the revolutions will vary directly, as the ratio* $\left(\frac{v_2^{x_2}}{v_1^{x_1}}\right)$ *, where* x_2 *and* x_1 *depend upon the ratios of the actual speeds* v_1 *and* v_2 *to the Chart Condition Speed V, at any other condition of resistance, so long as it is operating in the wake of the same hull, provided the effective thrusts do not exceed the point of cavitation.*

CAVITATION.

At one time the author advanced the statement that the point of cavitation depended upon tip speed and effective thrust. He now considers that this statement is in error and that the words *tip speed and* should be eliminated, for the tip speed at cavitation will vary with the effective horsepower being delivered while for equal effective thrusts this effective horsepower will vary inversely as the speed of ship since

$$\text{effective thrust} = \frac{\text{effective horsepower} \times 33{,}000}{\text{speed of ship} \times 101.33},$$

and a change in effective horsepower carries with it a change in revolutions. Therefore:

RULE 6.—*Should a propeller working in the wake of a certain hull cavitate at a certain speed while delivering a certain*

effective horsepower, it will, should the loading of the ship be changed, cavitate at any new speed where the effective thrust delivered is the same as that being produced under the original conditions of loading.

These cavitating thrusts will, under Chart 5, "Screw Propeller," conditions occur at about 10 per cent. increase over the effective thrusts given on that chart.

Therefore, having under chart conditions

$$\text{E.T.}_{\text{p}} = \frac{\text{E.H.P.} \times 33{,}000}{\text{V} \times 101.33 \times 36\pi \text{D}^2},$$

the value of effective thrust per square inch of projected area at the approximate point of cavitation will be

$$\text{E.T.}_{\text{p cav}} = \frac{1.10 \times \text{E.H.P.} \times 33{,}000}{\text{V} \times 101.33 \times 36\pi \text{D}^2}.$$

Where k_1 exceeds unity, due to decrease in density of the column of water flowing to and through the propeller, E.T._{p} and $\text{E.T.}_{\text{p cav}}$, become, respectively

$$\text{E.T.}_{\text{p}k_1} = \frac{\text{E.T.}_{\text{p}}}{k_1^3} = \frac{\text{E.H.P.} \times 33{,}000}{k_1^3 \times \text{V} \times 101.33 \times 36\pi \text{D}^2}, \text{ and}$$

$$\text{E.T.}_{\text{p}k_1 \text{ cav}} = \frac{\text{E.T.}_{\text{p cav}}}{k_1^6} = \frac{1.10 \times \text{E.H.P.} \times 33{,}000}{k_1^3 \times \text{V} \times 101.33 \times 36\pi \text{D}^2}.$$

At any other speed, v, for which an effective horsepower, *e.h.p.*, is required, the propeller will, therefore, be cavitating or on the verge of cavitation when

$$e.t._{\text{p}} = \frac{e.h.p. \times 33{,}000}{k_1^3 \times v \times 101.33 \times 36\pi \text{D}^2} =$$

$$\text{E.T.}_{\text{p}k_1 \text{ cav}} = \frac{1.10 \times \text{E.H.P.} \times 33{,}000}{k_1^3 \times \text{V} \times 101.33 \times 36\pi \text{D}^2}$$

and this explains why with slow-speed vessels having high values of k_1 cavitation occurs at such low values of effective thrust.

POINTS TO BE CONSIDERED IN PROPELLER DESIGN AND ANALYSIS.

Before taking up the actual work of analyzing or designing propellers to illustrate the foregoing, it is necessary to fully consider and bear in mind the following points:—

(1) The effective horsepower that will be delivered by any propeller with any application of power to the propeller shaft, depends upon the actual projected area ratio of the propeller, but does not depend upon the distribution of the projected area, in other words, does not depend upon the shape of the projection.

(2) Revolutions are affected by shape of projection; thus a broad-tipped blade or one with a bulbous form, will, with equal applied powers, turn slower than a blade having the standard form of projection (Chart 8, Sheet 24, "Screw Propellers"), but the thrusts will be greater, while the effect of narrowing the tip of the projection will be just the opposite.

(3) When propellers are working so close to the hull as to influence the effect of K, broad or bulbous-tipped blades will tend to increase K above that due to standard formed projections, while to narrow the tip below the standard will hold down this increase, while k_1 will apparently be affected inversely.

(4) Revolutions depend upon the rate of flow and the density of water flowing to the propeller, and are therefore affected by the value of k_1; that is, as the indicated or shaft horsepower necessary to deliver any effective horsepower increases due to an increase in k_1, the revolutions increase accordingly.

(5) The hull has no effect upon the effective horsepower that can be delivered by any given propeller except in so far as it affects the value of k_1.

(6) The hull increases or decreases the indicated or shaft horsepower, depending upon its effect upon the factor k_2.

(7) The hull affects the revolutions, as the value of V for the Chart Condition increases with the slip block coefficient.

(8) In comparing estimate of with the actual performances the difference between model-tank conditions of hull smoothness and water conditions and those that actually exist under the trial conditions of the real ship must be appreciated. It is hardly possible to obtain an exact similarity of these conditions, and therefore the actual *e.h.p.* for the different speeds and those given by the model tank for these same speeds may differ to an appreciable degree, and a considerable difference between the estimate and the actual performance will thus be produced.

The principal causes of such differences of condition are: Roughness of ship's bottom; adverse condition of wind and sea; undue shallowness of water on trial course; poor helmsmanship; possible error in assuming that the "Law of Comparison" holds for the hull appendages.

On this latter point, however, the estimates made where trial conditions were good in all respects, agree so closely with the actual performances that it appears practically certain that

THE LAW OF COMPARISON DOES APPLY TO THE HULL APPENDAGES.

(9) Another point which must be taken into consideration is the effect of various blade sections on the efficiency of performance of the propeller.

Where different blade sections are used for several propellers having the same diameter, same form and amount of projected area, and same real (not nominal) pitch, that propeller which has blade sections offering the least resistance to motion through the water will have the greatest efficiency.

By real pitch is meant the actual pitch of the propeller as modified by the influence of the back of the blade. By nominal pitch is meant the pitch of the helical pitch surface and

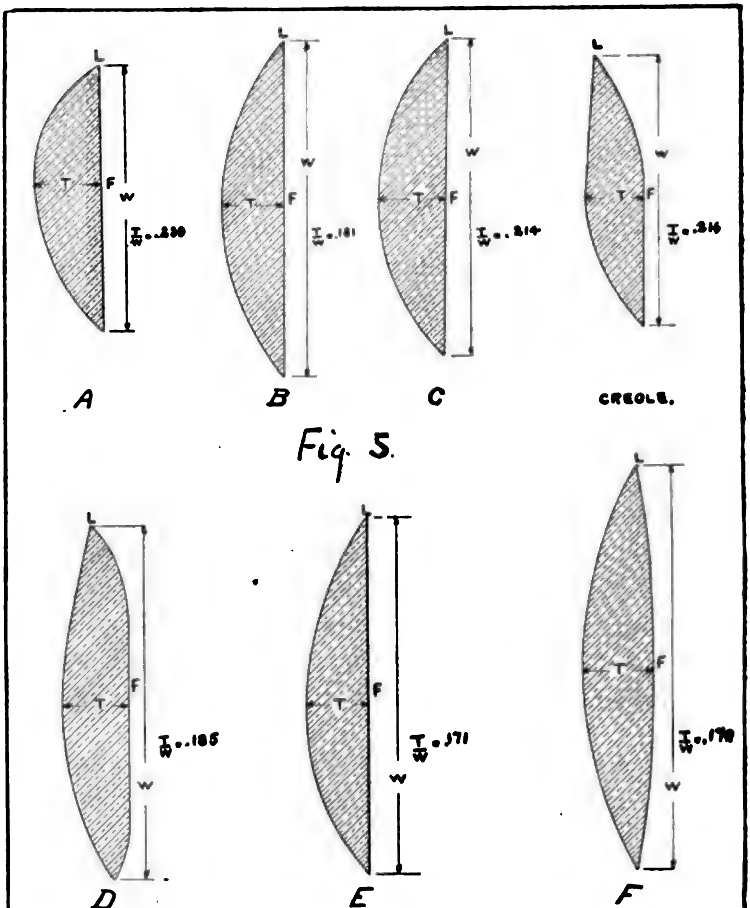


Fig. 5.

SECTIONS AT ROOT OF BLADE.

is the pitch referred to when the pitch of a propeller is spoken of.

The propellers from which the values of z were obtained were all of manganese-bronze or Monel metal; blades were all machined to pitch with thicknesses brought down to template; blade bolts, where propellers were of the built-up variety, carefully recessed into hub and fairing covering plates fitted over them; propeller blades and hubs highly polished.

The blade sections were all nearly similar to those marked B and C, Fig. 5; where differences existed, they were but slight.

Should the blades be very wide and very thin as compared with their width, variation of form of section has little effect. Should, however, the blades be narrow, the thickness ratio increases very greatly and blade section becomes of great importance. Thus a section similar to those marked Creole and D, Fig. 5, produce abnormal increases in resistance and the indicated horsepower per revolution becomes increased very considerably above that required for Sections B and C under similar conditions of hull resistance.

A modification of the section marked Creole, in which the leading edge was thrown back from the pitch face about ½ the blade thickness, and the face itself from the medium line was gradually curved back to this edge, produced an increase in k_1 of the total figure .02, such that k_1, if with the standard section equalled 1.02, with the modified section would equal $1.02 + .02 = 1.04$, and $K = (k_1 + .02 + k_2)$. This increase in k_1 is, however, not actual but is due to the blade section which causes an undue increase in real pitch, which in turn produces an increase in $P \times R$, I.H.P., E.H.P. and V over those given by the charts.

The section marked F tends to bring the real pitch more nearly equal to the nominal pitch of the blade than does the standard sections B and C, and the blades having this section would turn up faster than blades of the same nominal pitch but of standard section, due to this one fact alone. They possibly do have slightly less resistance to motion through the water than do those of standard section, but this cannot be stated positively.

The writer only uses Section F where the material of the designed propeller is to be much weaker than the standard material, manganese-bronze. In such cases the thickness is computed both for manganese-bronze and for the weaker material, and the difference in thickness resulting is added on to the face of the blade.

ILLUSTRATIONS OF THE FOREGOING.

In illustrating the use of the curves of z, w and K, the following vessels have been taken:

1. Destroyer.—Two-shaft, reduction-gear drive. Trials run in deep water. Single strut on each shaft.
2. Destroyer.—Two-shaft, turbine-driven. Trials run in deep water. Two struts on each shaft.
3. Battleship.—Four-shaft, turbine-driven. Trials run in deep water.
4. Battleship.—Two-shaft, reciprocating engine. Sister ship to 3. Trials run in deep water.
5. Battleship.—Four-shaft, turbine-driven. Trials run in deep water.
6. Battleship.—Sister ship to 5. Different propellers. Increased value of K. Trials run in deep water.
7. Battleship.—Two-shaft, reciprocating engines. Trials run in deep water.
8. Collier.—Two-shaft, electric-driven. Trials run in deep water.
9. Gunboat.—Single-screw, reciprocating engine.

The propellers of all the above vessels are three-bladed.

In all the above cases where S.H.P. was measured by means of torsion meters, the Gary-Cummings torsion meter was used and the results obtained speak volumes in favor of the accuracy of that instrument.

In only one or two of the illustrations will the estimate of revolutions be made, as these will be sufficient to indicate the method of computation.

Case 1.—Destroyer. Slip block coefficient = .385.

P.A. ÷ D.A. = .55, 3-bladed propeller, standard form.

$P = 8.636$ feet.

$D = 7.625$.

T.S. = 11,800 (Fig. 2).

$R = \text{T.S.} \div \pi D = 492.6$; $P \times R = \log 3.62880$.

$\text{I.T.}_D = 10.27$ (Fig. 2).

$$\text{I.H.P.} = \frac{2 \times D^2 \times \text{I.T.}_D \times P \times R}{291.8} = 17{,}410.$$

Mean T.C. very large.

P.C. = .5325 ∴ E.H.P. = 9,270, S.H.P. = 16,017.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. S.H.P.$_d$	Actual S.H.P.	$\frac{v}{V}$	x	Est. R_d.	Act'l R_d.
.025	232	10.8	1.603	1	399	400	..3263	1.18	131.4	130.5
.05	464	13.5	1.359	1	700	700	.4078	1.15	175.6	175.
.075	696	15.3	1.165	1	1,095	1,030	.4622	1.18	191.8	197.
.1	927	16.8	1.0268	1	1,506	1,510	.5075	1.21	216.8	217.
.2	1,854	20.4	.728	1	2,996	3,000	.6163	1.26	267.6	268.
.3	2,781	22.4	.5493	1	4,522	4,500	.6767	1.25	302.4	301.25
.4	3,708	23.8	.4238	1	6,037	5.900	.7190	1.25	326.2	326.25
.5	4,635	25.2	.3266	1	7,551	7,600	.7596	1.23	352.2	352.5
.6	5,562	26.3	.2432	1	9,149	9,200	.7946	1.21	374.2	372.5
.7	6,489	27.5	.1667	1	10,912	10,900	.8308	1.21	393.6	393.75
.8	7,416	28.7	.1105	1	12,419	12,650	.8671	1.18	416.3	416.25
.9	8,343	29.8	.0540	1	14,144	14,400	.9003	1.14	437.0	438.0
1.0	9,270	30.8	.0000	1	16,017	16,300	.9305	3.80	465.0	465.0

$\log R_d = \{\log R + 3 \log k_1 + x (\log v - \log V)\}$, where, in this case, $k_1 = 1.00$.

Case 2.—Number of Propellers = 2. Blades = 3.

P.A. ÷ D.A. = .607 R = T.S. ÷ πD = 600.5

P = 6′.5 P × R = log 3.59146

D = 7.5 I.T.$_D$ = 12.23

T.S. = 14,150

I.H.P. = 18,410, P.C. = .5226, E.H.P. = 9,616, S.H.P. = 16,935.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. S.H.P.$_d$	Actual S.H.P.$_d$
.025	240	10	1.603	1	422	440
.05	481	12.7	1.395	1	741	830
.075	723	14.5	1.165	1	1,158	1,220
.1	962	15.9	1.0268	1	1,592	1,600
.2	1,924	19.6	.7280	1	3,183	3,130
.3	2,885	21.85	.5493	1	4,781	4,750
.4	3,846	23.31	.4238	1	6,383	6,300
.5	4,808	24.55	.3266	1	7,965	7,780
.6	5.770	25.71	.2432	1	9,674	9,400
.7	6,731	26.81	.1667	1	11,537	11,260
.8	7,693	27.85	.1105	1	13,131	13,200
.9	8,654	28.82	.0540	1	14,955	14,950
1.0	9,616	29.90	.0000	1	16,935	16,750

Mean T.C. very large.

Case 3.—Slip Block Coefficient = .610 (Mean for all Propellers). Number of Propellers = 4. No. of Blades = 3. Mean T.C. Inboard Prop. = $3'.33$.

$$\text{P.A.} \div \text{D.A.} = .558 \qquad \text{I.T.}_{\text{D}} = 10.6$$
$$\text{P} = 8'.5 \qquad \text{P.T.}_{\text{P}} = 9.95$$
$$\text{D} = 9.17 \qquad \text{E.T.}_{\text{P}} = 12.25$$
$$\text{T.S.} = 12{,}080 \qquad 1 - \text{S} = .8122$$
$$\text{R} = \text{T.S.} \div \pi\text{D} = 419.7 \qquad \text{V} = \frac{\text{P} \times \text{R} \times (1 - \text{S})}{101.33} = 28.4$$
$$\text{P} \times \text{R} = 3{,}568$$

I.H.P. = 43,596, P.C. = .527, E.H.P. = 22,976, S.H.P. = 40,108. $k_1 = 1.02$.

$\frac{e.h.p.}{\text{E.H.P.}}$	*e.h.p.*	*v*	*z*	K	Est. S.H.P.$_d$	Actual S.H.P.$_d$	$\frac{v}{\text{V}}$	*x*	R$_d$ Est.	Act. R$_d$
.1	2,298	12.1	1.0268	1.02	3,846	3,900	.426	1.12	171.27	173.5
.2	4,595	15.05	.7280	1.02	7,653	7,800	.53	1.12	218.8	219.
.3	6,893	17.	.5493	1.02	11,549	11,600	.5986	1.14	248.13	247.5
.4	9,190	18.8	.4238	1.02	15,418	15,600	.662	1.18	273.75	273.5
.5	11,488	19.81	.3266	1.02	19,286	19,350	.6975	1.16	293.3	292.5
.6	13,786	20.51	.2438	1.02	23,368	22,900	.7222	1.12	302.3	309.0
.7	16,083	21.08	.1667	1.02	27,870	27,000	.7423	1.05	325.7	325.0
.8	18,380	21.60	.1105	1.02	31,720	31,700	.7606	.99	339.7	330.0

Case 4.—Slip B.C. = .60. Two Propellers, 3 Bladed.

$$\text{P.A.} \div \text{D.A.} = .328 \qquad \text{P.T.}_{\text{P}} = 8.79$$
$$\text{P} = 19'.75 \qquad \text{E.T.}_{\text{P}} = 10.28$$
$$\text{D} = 18'.25 \qquad 1 - \text{S} = .8551$$
$$\text{T.S.} = 7{,}240 \qquad \text{V} = 21.04$$
$$\text{R} = 126.27 \qquad \text{I.H.P.} = 24{,}767$$
$$\text{P} \times \text{R} = 2{,}494 \qquad \text{P.C.} = .663$$
$$\text{I.T.}_{\text{D}} = 4.35 \qquad \text{E.H.P.} = 16{,}420$$

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K × I.H.P.d	Actual K × I.H.P.d
.075	1,232	9.5	1.165	1.02	1,728	1,600
.1	1,642	10.45	1.0268	1.02	2,373	2,200
.2	3,284	13.21	.7280	1.02	4,726	4,800
.3	4,926	15.05	.5493	1.02	7,131	7,125
.4	6,568	16.45	.4238	1.02	9,521	9,300
.5	8,210	17.7	.3266	1.02	11,909	11,600
.6	9,852	18.92	.2432	1.02	14,430	14,625
.7	11,494	19.76	.1667	1.02	17,209	17,200
.8	13,136	20.33	.1105	1.02	19,587	19,600
.9	14,778	20.8	.05402	1.02	22,307	22,200
1.0	16,420	21.24	.00000	1.02	25,262	25,200
1.05	17,230	21.44	.02200	1.02	26,575	26,600
1.10	18,062	21.62	.04400	1.02	27,975	28,100

Mean T.C. = 2′.88.

Blade Section a modified "Creole" producing increase in k_1 and K from 1.00 to 1.02.

Case 5.—Slip B.C. = .62. Four Propellers, 3 Bladed. Mean T.C. of inner screws very large = 3′.5. Mean value of K for wing and inner propellers = 1.00. Slip B.C. is the mean for position of wing and inner screws.

P.A. ÷ D.A. = .501 | I.T.$_D$ = 8.75
P = 8′.193 | I.H.P. = 32,031
D = 9.588 | P.C. = .559
T.S. = 10,680 | E.H.P. = 17,905
R = 354.6 | S.H.P. = 29,469
P × R = 2,905

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Estimated S.H.P.d	Actual S.H.P.d
.1	1,791	10.35	1.0268	1	2,771	2,750
.2	3,581	13.05	.7280	1	5,513	5,500
.3	5,372	14.85	.5493	1	8,319	8,200
.4	7,162	16.35	.4238	1	11,106	11,150
.5	8,953	17.51	.3266	1	13,892	13,900
.6	10,744	18.52	.2432	1	16,833	16,600
.7	12,534	19.41	.1667	1	20,076	19,400
.8	14,324	20.2	.1105	1	22,849	22,500
.9	16,116	20.76	.05402	1	26,022	25,700
1.0	17,905	21.24	.00000	1	29,469	29,469
1.05	18,084	21.27	.02200	1	31,000	31,000

This estimate and the actual S.H.P.$_d$ are in error, for by checking the value of k_1 by means of the actual revolutions, it is found to be 1.025, which corresponds to the minimum value of K and also the value of K for the actual mean T.C. of the vessel. The error in actual results was caused by calibrating a full length of shafting, which included an enlarged portion for a spring bearing. A short section only of this shaft was included in the torsion-meter length and this short section included the enlarged bearing.

The result was that the torsion-meter readings and the resultant powers were too small. The error also showed up in the measured steam per S.H.P., which indicated, on the basis of this power, extremely uneconomical turbines.

Case 6.—Sister ship to Case 5. 4 Propellers—3 Blades.

Mean T.C. Wing screws = 2.24. K (Wing) = 1.10.
Mean T.C. Inner screws = 2.24. K (Inner) = 1.10.
Mean K = 1.1.

P.A. ÷ D.A. = 523 P × R = 2,893
P = 8′.188 I.T.$_D$ = 9.4
D = 10′ I.H.P. = 37,192
T.S. = 11,100 P.C. = .547
R = 353.4 E.H.P. = 20,392
S.H.P. = 34,296

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	K	Estimated K × S.H.P.$_d$	Actual K × S.H.P.$_a$
.1	2,039	11.05	1.10	3,547	3,700
.2	4,078	13.9	1.10	7,057	7,100
.3	6,118	15.81	1.10	10,650	10,650
.4	8,156	17.25	1.10	14,218	14,800
.5	10,196	18.53	1.10	17,784	19,100
.6	12,236	19.61	1.10	21,549	22,600
.7	14,274	20.40	1.10	25,700	25,500
.8	16,313	21.	1.10	29,250	29,200
.9	18,353	21.46	1.10	33,313	33,500

Case 7.—Slip B.C. = .655. Two Propellers, 3 Bladed. Mean T.C. = 2'.25.

P.A. ÷ D.A. = .308   P × R = 2,289
P = 18' $I.T._D = 4$
D = 17'.25   I.H.P. = 18,670
T.S. = 6,890   P.C. = .672
R = 127.14   E.H.P. = 12,546

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. $K \times I.H.P._d$	Actual $K \times I.H.P._d$
.1	1,255	9.85	1.0268	1.15	2,018	......
.2	2,509	12.45	.728	1.15	4,016	......
.3	3,764	14.02	.5493	1.15	6,061	......
.4	5,018	15.18	.4238	1.15	8,092	7,900
.5	6,273	16.05	.3266	1.15	10,121	9,750
.6	7,528	16.85	.2432	1.15	12,264	11,950
.7	8,782	17.58	.1667	1.15	14,626	14,250
.8	10,037	18.19	.1105	1.15	16,647	16,600
.9	11,291	18.62	.05402	1.15	18,959	18,750

Case 8.—Slip B.C. = .627. Two Propellers, 3 Blades. Mean T.C. = 1'.6. K = 1.22. Vessel under Rule 3.

P.A. ÷ D.A. = .304 $I.T._D = 3.85$
P = 14'.436   I.H.P. = 13,060
D = 15'.95   P.C. = .675
T.S. = 6,740   E.H.P. = 8,815
R = 134.4   S.H.P. = 12,020.
P × R = 1,943

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. $K \times S.H.P._d$	Actual $K \times S.H.P._d$
.1	882	8.77	1.0268	1.22	1,379	1,500
.2	1,763	11.18	.728	1.22	2,743	2,780
.3	2,645	12.77	.5493	1.22	4,140	4,160
.4	3,526	14.0	.4238	1.22	5,527	5,460
.5	4,408	14.95	.3266	1.22	6,913	7,140

Case 9.—Gunboat. Single Screw. Slip B.C. $=.805$. Trial displacement 2 per cent. below that corresponding to model. Trials run with and against tide, one run each way for trial points. Mean T.C. $= 2'.75$ ∴ Minimum value of K.

P.A. ÷ D.A. $= .243$	P.T.$_P = 7.57$
$P = 10'.625$	E.T.$_P = 8.29$
$D = 10'$	$1 - S. = .9132$
T.S. $= 5,350$	$V = 16.31$
$R = 170.3$	I.H.P. $= 1,637$
$P \times R = 1,809$	P.C. $= .694$
I.T.$_D = 2.64$	E.H.P. $= 1,136$

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	*v*	*z*	K	Est. K×IHP$_d$	Act. K×IHP$_d$	$\frac{v}{V}$	*x*	k_1	Revs. Est.	Revs. Act.
.1	114	8.1	1.0268	1.39	214	...	.4968	1.18	1.05	...	...
.2	227	10.1	.728	1.39	426	475	.6194	1.24	1.05	108.9	109
.3	341	11.18	.5493	1.39	642	612	.6856	1.28	1.05	121.6	124
.4	454	11.99	.4238	1.39	862	760	.7353	1.31	1.05	131.8	134.9
.5	568	12.62	.3266	1.39	1,070	950	.7738	1.28	1.05	142.0	145.8

Performances of Vessels with Four-Bladed Propellers.

It is now pertinent to pass to vessels having propellers fitted with four blades and to show that these propellers check equally as well as the three-bladed ones, when certain corrections for change in efficiency have been made.

The chart condition of performance of such wheels is that for three-bladed wheels having only 3/4 the projected area ratio, the chart I.H.P. being increased over that for the three-bladed wheel by the factor 1/.865, while for a two-bladed wheel the analysis would be for a three-bladed wheel of 3/2 the actual projected area ratio, while the I.H.P. factor would become .75.

The efficiency of a propeller, however, depends upon its total projected area ratio, therefore, in the analysis of the wheel, as also in the design, the propulsive coefficient to use is the chart propulsive coefficient (Fig. 2) corresponding to the actual projected area ratio and not to the ratio of the basic three-bladed propeller.

In illustrating this point, four vessels will be given, as follows:

Case 10.—Sister ship to Case 7. Propellers of same diameter as that vessel. Twin screws.

Case 11.—Vessel of merchant type. Single screw.

Case 12.—Vessel of gunboat type. Single screw.

Case 13.—Vessel of gunboat type. Single screw.

Case 10.—Slip B.C. of vessel = .655. Tip clearance same as Case 7 $\therefore$ K = 1.13. k_1 = 1.03, same as Case 7. P.A. ÷ D.A. (4 blades) = .391. 3 blades = .293.

P = 17′.8125	E.T.$_p$ = 9.53
D. = 17′.25	I − S = .884
T.S. = 6,525	P × R = 2,145
R = 120.4	V = 18.66
I.T.$_p$ = 3.63	I.H.P. = 18,380
P.T.$_p$ = 8.4	P.C. = .624
	E.H.P. = 11,476

e.h.p. / E.H.P.	*e.h.p.*	*v*	*z*	K	Est. K×IHP$_d$	Act. K×IHP$_d$	$\frac{v}{V}$	*x*	k_1	Revs. Est.	Revs. Act.
.1	1,148	9.4	1.0268	1.15	1,987	2,000	.5037	1.19	1.02	56.49	54.5
.2	2,295	11.75	.728	1.15	3,954	3,850	.6154	1.23	1.02	72.31	69.
.3	3,443	13.4	.5493	1.15	5,967	5,940	.7181	1.42	1.02	79.81	80.
.4	4,590	14.6	.4238	1.15	7,966	8,080	.7824	1.50	1.02	88.39	88.4
.5	5,738	15.43	.3266	1.15	9,965	9,950	.8269	1.60	1.02	94.45	94.5
.6	6,886	16.15	.2432	1.15	12,074	11,850	.8655	1.55	1.02	102.1	100.
.7	8,033	16.83	.1667	1.15	14,400	14,200	.902	1.68	1.02	107.4	105.75
.8	9,181	17.52	.1105	1.15	16,351	16,750	.9389	2.15	1.02	111.5	111.5
.9	10,328	18.1	.05402	1.15	18,666	19,000	.970	3.4	1.02	115.2	116.

The effective horsepowers here are estimated by adding 7 per cent. to the model-tank effective horsepower curves for a displacement of 16,000 tons, the actual displacement being 17,400 tons.

Case 11.—Vessel of merchant type, 6,200 tons displacement. Slip B.C. = .66. Single Screw. Reduction Gear. Power measured forward of gear and of thrust bearing. Estimated loss through gear and bearing estimated at 4 per cent.

Exceptionally fine aft and propellers placed very low. P.A. ÷ D.A. (4 blades) = .3497. 3 blades = .2621.

$P = 16'$	$E.T._p = 8.92$
$D = 15.5$	$1 - S = .887$
$T.S. = 5{,}820$	$V = 16.76$
$R = 119.5$	$I.H.P. = 5{,}497$
$P \times R = 1{,}912$	$P.C. = .65$
$I.T._D = 3.02$	$E.H.P = 3{,}573$
$I.T._D = 7.92$	

Mean T.C. = 4′ ∴ K = 1.125. Shallow water (20 fathom) course.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. $K \times SHP_d$	Act. $K \times SHP_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Revs. Act.
.1	357	8.0	1.0268	1.125	581	542	.4833	1.16	1.03	56.04	55.3
.2	714	10.2	.728	1.125	1,137	1,200	.6205	1.24	1.03	72.07	71.75
.3	1,072	11.70	.5493	1.125	1,746	1,834	.713	1.35	1.03	82.5	82.5
.4	1,429	12.80	.4238	1.125	2,331	2,424	.7799	1.49	1.03	89.92	91.0
.5	1,787	13.75	.3266	1.125	2,915	3,096	.8383	1.66	1.03	97.2	98.3
.6	2,143	14.55	.2432	1.125	3,532	3,686	.8843	1.79	1.03	104.5	104.0
.7	2,501	15.3	.1667	1.125	4,213	4,124	.9274	2.27	1.03	109.9	109.0

The excess actual $K \times S.H.P._d$ over the estimate may, in this case, be caused by the effect of the shallow water in the course, as will be shown later, but this is doubtful.

Case 12.—Gunboat. Single Screw. Slip B.C. = 702. Mean T.C. = 2′.0. K =

P.A. ÷ D.A. (4 Blades) = 4. 3 Blades = 3.

$P = 11'.5$	$P.T._p = 8.48$
$D = 9'.67$	$E.T._p = 9.55$
$T.S. = 6{,}670$	$1 - S = .8873$
$R = 219.6$	$V = 22.12$
$P \times R = 2{,}525$	$I.H.P. = 3{,}517$
$I.T_D = 3.76$	$P.C. = .62$
$E.H.P. = 2{,}181$	

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×IHP$_d$	Actual K×IHP$_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Act.
.1	218	9.75	1.0268	1.175	389	408	.4408	1.07	1.012	94.73	96.
.2	436	11.82	.728	1.175	747	780	.5344	1.06	1.012	117.1	117.8
.25	545	12.65	.630	1.175	970	990	.5719	.96	1.012	133.1	128.

Case 13.—Gunboat. Single Screw. Sister to Case 9. Displacement 6 per cent. below that corresponding to model. No standardization. Full-speed trial at 13.17 knots, held over outside course, one run north and one south. Result not reliable. Mean referred T.C. = 2.83 ft. P.A. ÷ D.A. (4 blades) = .338. 3 blades = .254.

$$
\begin{array}{rlrl}
P &= 11' & E.T._p &= 8.51 \\
D &= 9'.5 & 1 - S &= .9131 \\
T.S. &= 5{,}630 & V &= 18.7 \\
R &= 188.6 & I.H.P. &= 2{,}114 \\
P \times R &= 2{,}075 & P.C. &= .657 \\
I.T._D &= 2.85 & E.H.P. &= 1{,}389 \\
P.T._p &= 7.77 & &
\end{array}
$$

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×IHP$_d$	Act. K×IHP$_d$	$\frac{v}{V}$	x	k_1	Revs. Est.	Act.
.1	139	8.62	1.0268	1.39	277	...	.461	1.1	1.05	93.18	...
.2	278	10.64	.728	1.39	550	...	.5691	1.13	1.05	112.9	...
.3	417	11.73	.5493	1.39	830	...	.6274	1.12	1.05	129.6	...
.4	556	12.53	.4238	1.39	1,108	...	.6702	1.08	1.05	141.7	...
.5	694	13.17	.3266	1.39	1,385	1,208	.7044	1.02	1.05	152.8	146.7

EFFECT OF INCREASE OF LOAD FOR CONSTANT SPEED ON PROPELLERS HAVING DIFFERENT DESIGN CONDITIONS.

It has often been noted in the cases of large vessels driven by turbines and by reciprocating engines, that when the resistance for any given speed was increased to any great extent, the propellers of the reciprocating engined vessel immediately

demonstrated a great superiority over those of the turbine engined one.

This superiority depends upon the ratios $\frac{e.h.p.}{E.H.P.}$ and $\frac{v}{V}$. Where $\frac{e.h.p.}{E.H.P.} = 1$, and $\frac{v}{V}$ is less than unity, the effect of load increase for any given speed is much greater than when $\frac{e.h.p.}{E.H.P.} = 1$, $\frac{v}{V} = 1$, or when $\frac{e.h.p.}{E.H.P.} = 1$, and $\frac{v}{V} > 1$. The lower the value of $\frac{v}{V}$ for $\frac{e.h.p.}{E.H.P.} = 1$, the greater will be the increase in power percentage for the given speed.

This will be illustrated by a comparison of the effect of increase in load over trial load, for any given speed for the vessels of cases 3, 4, 5 and 6 given in the foregoing.

Suppose that these vessels have just completed their preliminary acceptance trials and after being commissioned are loaded down to their full capacity with stores, fuel, ammunition and reserve feed water preparatory to a cruise during which the standard speed will be 12 knots. Required the I.H.P. or S.H.P. and revolutions for this speed under the new conditions.

The *e.h.p.*'s for 12 knots at trial condition are: Case 3, 2,250; case 4, 2,460; case 5, 2,750: case 6, 2,650.

Assuming that when fully loaded, the vessels have an increase of 10 per cent. in displacement over their designed trial conditions, the *e.h.ps.* required for these increased displacements at 12 knots become:

Case 3, 2,350; Case 4, 2,568; Case 5, 2,880; Case 6, 2,800. Analyzing for the two conditions there results as follows:

Data for Analysis.

	Case 3.	*Case 4.*	*Case 5.*	*Case 6.*
V	28.4	21.04	24.03	23.75
E.H.P	22,976	16,420	17,905	20,392
I.H.P. or S.H.P	40,108	24,767	29,469	34,296
v	12	12	12	12
e.h.p. (light)	2,250	2,460	2,750	2,650
e.h.p. (heavy)	2,350	2,568	2,880	2,800
S	.1879	.1449	.1618	.1687
$\frac{v}{V}$	.423	.57	.499	.505
$\left(\frac{e.h.p.}{E.H.P.}\right)$ (light)	.098	.1498	.153	.13
$\left(\frac{e.p.h.}{E.H.P.}\right)$ (heavy)	.103	.1560	.161	.137
z light	1.029	.846	.837	.905
z heavy	1.005	.829	.815	.883
K	1.02	1.02	1.025	1.10
k_1	1.02	1.02	1.025	1.025
I.H.P.$_d$ or S.H.P.$_d$ (light)	3,751	3,531	4,289	4,268
K (I.H.P.$_d$ or S.H.P.$_d$) (light)	3,827	3,601	4,396	4,695
I.H.P.$_d$ or S.H.P.$_d$ (heavy)	3,965	3,672	4,512	4,490
K (I.H.P.$_d$ or S.H.P.$_d$) (heavy)	4,044	3,745	4,625	4,939
x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.}$ (light)	1.05	1.26	1.11	1.15
x for $\frac{v}{V}$ and $\frac{e.h.p.}{E.H.P.}$ (heavy)	.98	1.22	1.09	1.13
R	419.7	126.27	354.6	353.4
R$_d$ (light)	172.5	66.0	176.1	173.3
R$_d$ (heavy)	180.6	67.49	178.6	176.0
Per cent. increase power	5.7	4.0	5.2	5.2
Per cent. increase revolutions	4.7	2.3	1.4	1.4
$\frac{v}{V}$ for $\frac{e.h.p.}{E.H.P.} = 1$	.796	1.0095	.886	.92
(Light) actual K (I.H.P.$_d$ or S.H.P.)	3,800	3,600	4,300	4,800
(Light) actual R$_d$	172	66.6	176	171

It is here shown how that, immediately upon being put into cruising trim, all four vessels become more costly to drive than was shown on their trials, the reciprocating ship requiring 4 per cent. more power, while the turbine ships require an increase of 5.2, 5.2, and 5.7 per cent. increase respectively.

Now, suppose that immediately upon putting to sea they encounter such conditions of wind and sea that the effective horsepower required for each vessel to maintain the standard

speed of 12 knots, is increased by 500, then the increases over the trial conditions are:

	Case 3.	*Case 4.*	*Case 5.*	*Case 6.*
e.h.p.	2,850	3,068	3,380	3,300
e.h.p. ÷ E.H.P	.124	.187	.189	.162
z	.925	.753	.748	.82
I.H.P.$_d$ or S.H.P.$_d$	4,767	4,374	5,265	5,191
K	1.02	1.02	1.025	1.10
k_1	1.02	1.02	1.025	1.025
K × (I.H.P.$_d$ or S.H.P.$_d$)	4,862	4,462	5,396	5,710
$\frac{v}{V}$	.423	.57	.499	.505
x	.85	1.18	1.02	1.07
R$_d$	202.0	69.05	187.9	183.2
Per ct. increase of R$_d$	17.1	4.6	6.7	5.6
Per ct. increase of K (I.H.P.$_d$ or S.H.P.$_d$)	27	24	23	22

The small propeller of Case 3 is seen to increase heavily in revolutions and power as compared with the larger propellers of the other cases.

EFFECT ON PROPULSIVE EFFICIENCY OF TRIMMING VESSELS OF CERTAIN AFTERBODY LINES DOWN BY THE STERN.

Naval Constructor D. W. Taylor, U. S. N., has demonstrated that with vessels of the ordinary battleship form, in fact, with models of Case 7, a decrease in resistance of about 1¾ per cent. occurs when the ship is trimmed approximately three feet by the stern. This percentage decrease does not constitute the only gain that occurs in such cases, for by so trimming the vessel there is a considerable increase in wake with a corresponding wake gain for the propeller, so that with the thrust deduction remaining constant, there results a large gain in propulsive efficiency, made up of the two factors, decreased resistance and wake gain. In giving examples of such cases the total gain will be treated as decrease in resistance.

Case 14.—Battleships A and B. Slip B. C. = .608. Twin screws. Vessels alike in all respects except trim. A trimmed on even keel. B trimmed 3 feet by stern. P.A. ÷ D.A. = .32 (3-bladed).

$$
\begin{array}{rlrl}
P &= 19'.99 & \text{I.T.}_D &= 4.22 \\
D &= 18'.65 & \text{I.H.P} &= 24{,}416 \\
\text{T.S.} &= 7{,}110 & \text{P.C.} &= .666 \\
R &= 121.4 & \text{E.H.P.} &= 16{,}264 \\
P \times R &= 2{,}427 & &
\end{array}
$$

Est. $z = \log \text{I.H.P.} + \log K - \log (K \times \text{I.H.P.}_d)$. Mean T.C. $= 3'.19$.

A								B					
$\frac{e.h.p.}{\text{E.H.P.}}$	*e.h.p.*	v	z	K	Est. K × I.H.P.$_d$	Actual K × I.H.P.$_d$	Actual K × I.H.P.$_d$	K	Est. z	$\frac{e.h.p.}{\text{E.HP.}}$	*e.h.p.*	$\frac{e.h.p.B}{e.h.p.A}$
.1	1,626	9.55	1.0268	1.06	2,433	2,400	......	1.06	...	...	...	......
.2	3,253	12.35	.728	1.06	4,841	4,900	4,200	1.06	.7897	.172	2,798	.862
.3	4,879	14.15	.5493	1.06	7,306	7,300	6,300	1.06	.6136	.258	4,196	.860
.4	6,506	15.55	.4238	1.06	9,754	9,800	8,600	1.06	.4785	.350	5,693	.875
.5	8,132	16.7	.3266	1 06	12,200	12,300	11,100	1.06	.3677	.450	7,319	.900
.6	9,758	17.75	.2432	1.06	14,784	14,900	13,500	1.06	.2827	.550	8,945	.9167
.7	11,385	18.66	.1667	1.06	17,631	17,350	16,000	1.06	.2089	.652	10,604	.9314
.8	13,011	19.45	.1105	1.06	20,067	20,000	18,500	1.06	.1458	.740	12,035	.9269
.9	14,638	20.1	.05402	1.06	22,854	22,700	20,900	1.06	.0928	.834	13,564	.9267
1.0	16,264	20.6	.00000	1.06	25,881	25,500	23,200	1.06	.0475	.898	14,605	.898
1.05	17,077	20.85	.0220	1.06	27,226	27,200	24,700	1.06	.0203	.97	15,776	.9238
1.10	17,890	21.05	.0440	1.06	28,640	28,400	26,000	1.06	.0020	.999	16,248	.9082

There is thus seen to be an average reduction in resistance combined with wake gain of 7 per cent., throwing out the three lower points where indicator errors and percentage speed errors would be a maximum, or a portion of the gain may possibly be due to a change in the value of K, the actual change occurring in the value of k_2, which has become negative and has more than eliminated the entire value of K.

Case 15.—Again, to show the results of trim with certain classes of vessels, take the case of a sister ship to Case 7. This sister was trimmed about three feet by the stern, and obtained her designed speed on about 800 I.H.P. less than the Case 7 vessel, and with three revolutions less. The propellers were alike in all respects.

Using the same hull and propeller data and the same Chart Conditions for Case 15 as for Case 7, and estimating the resistances for Case 15, they (the effective horsepowers at the

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same speeds as in Case 7) are found to be as shown in the following table:

Correcting the value of K to 1.13.

v	Actual K×I.H.P.d	K	Est. x	$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	$\frac{\text{Case 12}}{\text{Case 7}}$
15.18	7,300	1.13	.4609	.362	4,542	.905
16.05	9,250	1.13	.3581	460	5,771	.920
16.85	11,230	1.13	.2738	.560	7,026	.931
17.58	13,500	1.13	.1939	.670	8,406	.957
18.19	15,750	1.13	.1269	.780	9,786	.975
18.62	17,800	1.13	.0738	.870	10,915	.967

The gain is seen to increase here as the vessel passes from the low to the high speeds, as was the case in the previous vessel, the mean apparent decrease of resistance for the three lower speeds being 8.1 per cent. and for the three higher speeds 3.7 per cent., the total mean decrease being 5.85 per cent.

EFFECT OF SHOAL WATER ON THE PERFORMANCE OF PROPELLERS.

In discussing this subject, several examples will be taken; the first will be

Vessel C: Ran in deep water; in order to establish K.

2d. Vessel D: Sister ship to C, but with different propeller of slightly less diameter but with same shaft lines as C. Ran in twenty fathoms of water.

3d. Vessel E: Sister ship to Case 2. Ran in twenty fathoms of water.

4th. Vessel F: Of same type as E, but 25 feet shorter on the water line. Ran in 20 fathoms of water.

4th. Vessel G: Sister ship to Case 1 and Case 2, but propellers located differently from either Case 1, Case 2, or E. Ran in twenty fathoms of water.

With vessels running in shallow water, the effect on the

factor K is very apparent. At the low speeds K may become even less than unity and increases gradually in value until unity may become exceeded quite considerably. The effect of shallow water appears to depend, however, upon the type of vessel, heavy vessels being adversely affected through all speeds, while light vessels feel the adverse effects from 15 knots up to 26 knots. Single-screw vessels, however, appear to be unaffected up to at least 15 knots. The major part of this increase is probably due to increase of k_1, which in some cases has become so excessive as to cause very heavy vibration of the vessel. In one recent case, the side of the vessel was broken in by the water hammer caused by k_1.*

Vessel C.

Vessel C.—Slip Block Coefficient = .64.

P.A. ÷ D.A. = .315	P × R = 2,289
P = 18 feet	I.T.$_D$ = 4.09
D = 17.54 feet	I.H.P. = 19,647
T.S. = 6,990 feet	P.C. = .67
R = 127.2	E.H.P. = 13,163

Twin Screws, 3 Blades; V = 19.65.

To find K. Speed of ships 19 knots, Revs. = 121.

$$\frac{e.h.p.}{E.H.P.} = .8 \therefore e.h.p. = 10{,}530. \quad K \times I.H.P._d = 16{,}300.$$

v corresponding to $e.h.p. = 19$

$$\frac{v}{V} = .9669$$

$$x \text{ for } \frac{v}{V} \text{ and } \frac{e.h.p.}{E.H.P.} = 4.25$$

$$z \text{ for } \frac{e.h.p.}{E.H.P.} = .8 = .1105$$

$$\begin{aligned} \log I.H.P. &= 4.29330 \\ z &= .11050 \\ \hline \log I.H.P._d &= 4.18280 \\ \log (K \times I.H.P._d) &= 4.21219 \\ \hline \end{aligned}$$

$$\log K = 0.02939 \therefore K = 1.07$$

* It should be borne in mind that k_1 is the measure of the density of the water flowing through the propeller. As k_1 increases the density decreases.

To find k_1.

$$k_1^3\ R_d \text{ for 19 knots} = 121$$

$$\log R = 2.10449$$

$$x\left(\log \frac{v}{V}\right) = 9.93791$$

$$\log R_d = 2.04240$$

$$3 \log k_1 + \log R_d = 2.08279$$

$$3 \log k_1 = 0.04031 \therefore \log k_1 = 0.01344 \therefore$$

$$k_1 = 1.0314.$$

Estimate of Power and Revolutions.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	x	Est. $I.H.P._d$	Est. K	Estimated $K \times I.H.P._d$	Actual $K \times I.H.P._d$	$\frac{v}{V}$	x	Est. R_d	Act. R_d
.1	1,316	10.5	1.0268	1,847	1.07	1,977	2,000	.5344	1.26	63.37	63.
.2	2,632	12.98	.728	3,675	1.07	3,933	4,275	.6606	1.32	80.74	79.
.3	3,949	14.62	.5493	5,547	1.07	5,935	6,200	.744	1.47	90.37	89.5
.4	5,265	15.6	.4238	7,405	1.07	7,923	7,600	.7939	1.55	97.6	96.2
.5	6,586	16.8	.3266	9,262	1.07	9,910	9,900	.855	1.95	102.8	104.5
.6	7,898	17.75	.2432	11,223	1.07	12,009	12,300	.9033	2.25	111.04	111.5
.7	9,214	18.45	.1667	13,385	1.07	14,322	14,450	.939	2.76	117.3	116.8
.8	10,530	19.	.1105	15,234	1.07	16,300	16,300	.9669	4.25	121.	121.

It having been shown how, in deep water, the estimated power and revolutions correspond very closely to the actual, turn now to the sister ship, D, and see the effect of the shallow course.

$P.A. \div D.A. = .327$	$I.T._D = 4.35$
$P = 19$ feet.	$V = 22.84$
$D = 16.5$ feet.	$I.H.P. = 21,540$
$T.S. = 7,240$	$P.C. = .663$
$R = 139.67$	$E.H.P. = 14,281$

$P \times R = 2,654$ $K = 1.07$; $k_1 = 1.0314$, as before.

Twin Screws, 3 Blades.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	Est. $I.H.P._d$	K	Est. K × $I.H.P._d$	Actual K × $I.H.P._d$	$\frac{v}{V}$	x	Est. R_d	Act. R_d
.1	1,428	10.75	1.0268	2,025	1.07	2,167	2,580	.4707	1.13	65.4	65.4
.2	2,856	13.3	.728	4,030	1.07	4,312	4,800	.5823	1.16	81.84	81.7
.3	4,284	14.98	.5493	6,081	1.07	6,507	6,800	.6559	1.2	92.38	92.4
.4	5,712	16.27	.4238	8,118	1.07	8,687	8,900	.7167	1.24	101.4	100.7
.5	7,140	17.28	.3266	10,154	1.07	10,865	11,100	.7566	1.21	109.35	107.5
.6	8,569	18.11	.2432	12,304	1.07	13,165	13,600	.7929	1.20	118.7	114.5
.7	9,997	18.79	.1667	14,674	1.07	15,701	16,547	.8227	1.175	121.96	120.1

By inspection it is seen that up to a speed of about 15 knots, the estimated and actual revolutions agree, while the actual power exceeds the estimated, thus indicating an increase in the value of k_2. From 15 knots upwards, the actual revolutions grow gradually lesser than the estimated, while the actual power increases above the estimated. This indicates an increase in wake gain, reducing the value k_1, accompanied at the same time by an abnormal increase in the value of k_2, due to the increased quantity and velocity of the stream-line flow from the surface of the water toward the keel, increasing the brake action on the propeller. This increase in k_2 may have been accompanied by increased hull vibration, but, due to lack of information, it cannot be stated positively whether this was the case.

The cases of vessels E, F and G in shallow water.

These vessels were all of light draught and of high speed. Vessel F being of a smaller size than the other two, her case will be examined first, and then the other two vessels, sisters of Case 2 and Case 1, deep-water tried ships, respectively.

Vessel F.

Slip B.C. = .343. Two propellers, three blades. Usual deep water values of K and k_1 = unity.

P.A. ÷ D.A. = .587	I.T.$_D$ = 11.5
P = 6.17 feet	V = 29.12
D = 6.67 feet	I.H.P. = 13,512
T.S. = 13,100	P.C. = .5225
R = 625.5	E.H.P. = 7,061
P × R = 3,857	S.H.P. = 12,431

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×S.H.P.$_d$	Actual K×S.H.P.$_d$	$\frac{v}{V}$	x	k_1	Est. R$_d$	Act. R$_d$
.025	177	10.	1.603	1	310	300	.3434	1.17	1	171.	182
.05	354	12.6	1.359	1	500	660	.4327	1.21	1	227.	225
.075	531	14.25	1.195	1	793	950	.4893	1.23	1	259.7	257
.1	706	15.7	1.0268	1	1,169	1,270	.5392	1.26	1	292.5	282
.2	1,412	19.4	.7280	1	2,380	2,330	.6662	1.33	1	364.4	350
.3	2,118	20.95	.5493	1	3,509	3,000	.7194	1.37	1	398.4	382
.4	2,824	23.2	.4238	1	4,685	4,500	.7967	1.57	1	437.8	429
.5	3,530	24.6	.3266	1	5,860	5,800	.8448	1.75	1	465.6	463
.6	4,236	25.9	.2432	1	7,101	7,250	.8891	1.88	1	501.5	498
.7	4,942	27.05	.1667	1	8,469	8,625	.9289	2.3	1	528.	534
.8	5,648	28.15	.1105	1	9,639	10,000	.9667	4.1	1	544.4	559
.9	6,354	29.25	.05402	1	10,973	11,400	1.0045	—3.2*	1	616.6	616
1.0	7,061	30.41	.00000	1	12,431	13,200	1.0443	1.32	1	663.9	665

* Estimated.

Discarding the lowest three speeds on account of possibly inaccurate torsion-meter readings due to the small amount of torque, it will be noted that up to 25.9 knots, excluding the 20.95 knot point where an error in plotting undoubtedly exists, the powers agree very closely while the actual revolutions rapidly decrease and then increase as compared with the estimated ones. This phenomenon indicates a heavy wake gain, reducing the value of k_1, with an equal loss due to an increase in k_2, thus maintaining a constant value of K. After passing 25.9 knots, k_1 gradually increases and then decreases again, that is, the wake at this part of the career is actually negative, while k_2 continues to hold its high value to the end of high-speed runs. k_1 here may produce heavy hull vibrations at from 27 to 29 knots speed.

Vessels E and G.

To give a better idea of conditions on these two vessels as compared with the deep-water tried vessels of Cases 1 and 2,

the following table of propeller characteristics and locations is given:

	E	Case 2.	G	Case 1.
Diameter of propeller,	92.5″	90″	91.5″	92″
Pitch of propeller,	82″	78″	78″	104″
Projected area ratio,	.595	.607	.615	.55
Actual tip clearance,	16¾″	19″	17″	25¾″
Hub above base,	4′–1″	4′	3′–9½″	3′–7½″
Middle of hub, for'd stern post,	0′–0″	4′	7′–0″	9′–0″
Middle of hub, abaft strut,	27¾″	2′	2′–0″	3′–0″

The vessels were alike in all but the following respects: The dead wood was cut away in an arch abreast the propellers of G and forward of the propellers of E and Case 2, the after end of the arch coming down slightly and forming a skag. In Case 1, while the dead wood was cut away, the arch was missing, the raised section of the keel running aft to the stern post in a perfectly straight line. Also E and Case 2 carried two supporting struts for each propeller shaft while F and Case 1 were fitted with only one.

In the order of smoothness of running they ranked in the following order: 1, Case 1; 2, Case 2; 3, E; 4, G. Case 1 vibrated only after reaching very high speeds, and then only moderately. Case 2 began vibrating at about 20 knots but was no worse than Case 1 at 29 knots. E vibrated more heavily than either 1 or 2 at high speed, while G failed, vibration commencing at about 20 knots and becoming more and more violent as the speed and power increased until the trial was brought to a finish by the side plating of the hull rupturing opposite the blade tips.

The Chart Condition and estimated performances of E and G, as compared with the actual performances will now be shown.

Vessel E.

Chart Conditions: V = 28.73; S.H.P. = 16,845; E.H.P. = 9,568; R = 557.5.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	K	Est. $K \times S.H.P._d$	Actual $K \times S.H.P._d$	$\frac{v}{V}$	k_1	x	Est. R_d	Act. R_d.
.025	239	9.9	1	425	311	.3446	1	1.17	160.3	158
.05	476	12.	1	678	622	.4177	1	1.2	195.6	190
.075	717	14.3	1	1,075	1,100	.4977	1	1.24	234.7	231
.1	957	15.5	1	1,584	1,500	.5395	1	1.26	256.2	250
.2	1,914	19.35	1	3,151	3,000	.6735	1	1.35	327.0	314
.3	2,871	21.6	1	4,755	4,550	.7518	1	1.52	361.3	353
.4	3,828	22.9	1	6,349	5,800	.7971	1	1.58	389.6	382
.5	4,784	24.1	1	7,941	7,250	.8388	1	1.67	415.7	410
.6	5,742	25.2	1	9,622	8,900	.8771	1	1.68	447.3	440
.7	6,699	26.24	1	11,502	10,750	.9133	1	1.88	470.1	469
.8	7,656	27.25	1	13,061	12,800	.9485	1	2.5	488.5	500
.9	8,613	28.25	1	14,876	14,900	.9833	1	5.05	...	538
1.0	9,568	29.3	1	16,845	16,800	1.02	1	.985	568.4	575
1.05	10,044	30.375	1	17,720	17,400	1.057	1	1.32	600.	614

As in the case of vessel F, there is a noteworthy decrease in actual revolutions from those estimated, beginning at 15 knots. This decrease reaches a maximum at about 20 knots, as in Case F, when it begins to diminish, again disappearing at about 26 knots, thus agreeing again with Case F. Carrying out the agreement with Case F, the actual revolutions are found exceeding the estimated from the 26-knot point upward, thus indicating practically an exact reproduction of Case F conditions with Case E.

The only difference which is found between the performance characteristics of these light, fast, shallow-draught vessels and those of the heavy battleship, D, is that in the latter case the heavy loss caused by the abnormal increase in k_2 more than offsets the gain due to increased wake from 15 knots upwards, while below 15 knots, as in the cases of E and F, there is no change in wake and therefore no change in k_1, but there is quite a heavy increase in k_2.

Vessel G.

As stated already, this vessel is similar to Case 1, 2, and E, except in the small particulars previously noted, but the

propeller is quite differently located, as shown by the table. The propellers themselves, resemble very closely, those of Case 2, the differences being so slight as to preclude any possibility of their having been the cause of the vessel's failure, particularly when it is stated that the two vessels, G and Case 2, ran their trials on practically the same displacement. It appears that the cause of the failure of G can be ascribed to location principally, a possible secondary cause being the difference in the keel lines of Case 1 and G abreast of the propeller, these differences having been already given.

Should this assumption of the cause of failure be correct, the comparison of estimated and actual powers and revolutions will show the variations of k_1 or k_2, as shown in E and F, and these variations will occur within the same speed ranges, but the comparison will also show an abnormal increase in k_2 or k_1 due to the braking action of the malign downward hull and strut currents on the inboard tips of the propeller blades, or due to the decreased density of the water in the propeller race, thus producing an abnormal increase in the power without any corresponding return in thrust by this increase.

The characteristics of the propeller having been given in the table, it remains necessary to give the Chart Conditions, which are:

S.H.P. = 18,374; E.H.P. = 10,385; V = 30.51; R = 615.

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. K×SHP$_d$	Actual K×SHP$_d$	$\frac{v}{V}$	x	k_1	Est. R$_d$	Act. R$_d$
.1	1,039	16.25	1.0268	1	1,727	1,675	.5326	1.25	1	279.8	271.2
.2	2,078	20.25	.728	1	3,438	3,480	.6637	1.32	1	358.	343.0
.3	3,117	22.23	.5493	1	5,189	5,225	.7286	1.40	1	394.8	385.5
.4	4,154	23.72	.4238	1	6,926	7,250	.7775	1.47	1	424.8	415.0
.5	5,193	25.11	.3266	1	8,662	9,650	.823	1.52	1	457.4	467.0
.6	6,234	26.27	.2432	1	10,496	11,800	.861	1.52	1	489.8	509.0
.7	7,270	27.57	.1667	1	12,518	14,060	.9037	1.72	1	516.7	565.0
.8	8,308	28.73	.1105	1	14,247	16,200	.9417	2.25	1	537.2	602.5
.9	9,351	29.87	.5402	1	16,225	18,000	.979	4.25	1	562.1	643.0

The hull having been repaired, the strut section axes slightly altered, and the plating opposite the propeller tips heavily stiffened to withstand the water-hammer blows, the original propellers were removed and others of the following characteristics were installed:

P.A. ÷ D.A. = .603	$I.T._D$ = 12.05
P = 6.67 ft.	V = 28.89
D = 7.71 ft.	I.H.P. = 18,910
T.S. = 13,900	P.C. = .5225
R = 574	E.H.P. = 9,871
P × R = 3,851	S.H.P. = 17,397

These propellers differ but slightly from those of E, being 80 inches in pitch instead of 82 inches, and having a very slightly increased projected area ratio:

The estimated and actual performances were according to the following table:

$\frac{e.h.p.}{E.H.P.}$	*e.h.p.*	v	z	K	Est. K×SHP_d	Actual K×SHP_d	$\frac{v}{V}$	x	k_1	Est. R_d	Act. R_d
.1	987	15.97	1.0268	1	1,638	1,575	.5528	1.29	1	267.2	257.
.2	1,974	19.95	.728	1	3,255	3,225	.6906	1.4	1	341.8	323.5
.3	2.961	21.99	.5493	1	4,911	4,600	.7612	1.59	1	371.9	364.
.4	3,948	23.44	.4238	1	6,557	6,420	.8114	1.71	1	385.	400.
.5	4,936	24.79	.3266	1	8,201	8,675	.8581	2.01	1	422.	439.
.6	5,922	26.00	.2432	1	9,938	10,900	.9000	2.25	1	452.8	480.
.7	6,910	27.17	.1667	1	11,853	13,225	.9405	2.85	1	481.9	526.
.8	7,896	28.25	.1105	1	13,490	15,200	.9778	5.9	1	503.9	568.
.9	8,883	29.35	.5402	1	15,362	17,120	1.016	.95	1	582.7	620.
1.0	9,871	30.45	.0000	1	17.397	19,000	1.054	1.6	1	624.4	640.

The results obtained agreeing closely with those obtained with the original propellers, and there being practically no decrease in hull vibration, it became necessary to look in other directions for a panacea. As great a change in position as was possible with the existing arrangement of struts was decided upon, and after cutting about four inches off the for-

ward end of the hubs to permit of moving the propellers that amount forward on the shafts, the propellers were replaced.

The vessel was then restandardized over the measured mile and the reduction in powers over previous trials was phenomenal, while there was a very considerable reduction in the vibrations.

The estimated and actual results of these trials are shown in the following table:

$\frac{e.h.p.}{E.H.P.}$	$e.h.p.$	v	z	K	Est. $K \times SHP_d$	Actual. $K \times SHP_d$	$\frac{v}{V}$	x	k_1	Est. R_d	Act. R_d.
.1	987	15.97	1.0268	1	1,635	1,575	.5528	1.29	1	267.2	258
.2	1,974	19.95	.728	1	3,255	3,100	.6906	1.4	1	341.8	325.1
.3	2,961	21.99	.5493	1	4,911	4,550	.7612	1.59	1	371.9	370
.4	3,948	23.44	.4238	1	6,557	6,150	.8114	1.71	1	401.1	402
.5	4,936	24.79	.3266	1	8,201	8,150	.8581	2.01	1	422	422
.6	5,922	26.00	.2432	1	9,938	10,175	.9000	2.25	1	452.8	471
.7	6,910	27.17	.1667	1	11,853	12,225	.9405	2.85	1	481.9	506.
.8	7,896	28.25	.1105	1	13,490	14,100	.9778	5.9	1	503.9	544
.9	8,883	29.35	.5402	1	15,362	16,100	1.016	.95	1	582.7	584
1.0	9,871	30.45	.0000	1	17,397	18,300	1.054	1.6	1	624.4	630

Inspection of the above results indicate that practically the same conditions exist as in the case of E, being probably slightly modified by the different position of the propeller from that of E. There is an apparent wake gain with absence of k_2 up to 26 knots. Above that speed k_1 becomes negative, causing an increase in power and revolutions. Between 28.25 and 29.35 knots the negative wake disappears and k_1 returns to its normal value of unity, the value of k_2, however, continuing to increase as the speed increases. The most striking feature of this final performance is the great reduction in power, accompanied by a reduction in hull vibrations, over the power and vibrations of the other two trials, these reductions being attributed solely to a change of a few inches in the fore-and-aft location of the propeller.

Conclusions as to the effect of shoal water on Propeller Performances.

Considerations of the above results leads to the following conclusions: Should a vessel with winged out propellers be tried over a course having approximately 20 fathoms depth of water, for speeds lower than 26 knots the revolutions will be lowered. For light-draught vessels of narrow beam as compared with length and of fine block coefficient the powers for the speeds will also be reduced. For deep-draught vessels having from medium to full block coefficient, this decrease of revolutions will be accompanied by an increase in the powers required for the given speeds.

For speeds above 26 knots, the revolutions will increase unduly until a speed of between 28 and 29 knots is reached, when the revolutions again become normal. The powers for the speeds above 26 knots will, however, increase unduly as compared with those required in deep water for the same speeds.

With vessels having single screws located directly abaft the stern post, the effect of this same depth of water on the course appears to be nil, as shown by Cases 11 and 12.

Use of Figures 1, 2 and 4 in Design.

In the author's work on "Screw Propellers" there is described a method of designing propellers to be followed when the desired revolutions and diameter of the propeller are so small that the resultant tip speed is such as correspond to the tip speed for a propeller of much greater diameter than that allowed, and working, under Chart Conditions, at a much lower number of revolutions than those desired.

Such problems are called "Problems of Reduced Diameter." By the use of Figures 1, 2 and 4, in conjunction with Chart 5, "Screw Propellers," these problems can readily be solved, and Chart 6, "Screw Propellers," can be eliminated from consideration.

On Fig. 4 are shown two cross curves marked "Limit Line

of Design for x," and the intersections of these lines with the curves of x are the starting points for the actual work of design.

As an example to illustrate the method, the following problem will be solved:

Problem in design of Propeller.

Slip block coefficient of vessel = .627.
Vessel of Rule 3 Class for determination of K. K = 1.22; $k_1 = 1.02$.

The diameter of the propeller must not exceed 16 feet 6 inches nor the revolutions about 90 per minute when running at a speed of 14 knots per hour. Find pitch and projected area ratios of the propeller and I.H.P., when the total effective horsepower to be delivered equals 3,650, divided between two propellers.

Computation.

Base ....$e.h.p.$ =. 3,650 3,650 3,650 3,650 3,650

Assume, in order to find load point for maximum efficiency.

(1) $\frac{e.h.p.}{E.H.P.}$ =	.2	.3	.4	.5	.6
(2) E.H.P. = 3,650 ÷ $\frac{e.h.p.}{E.H.P.}$	18,250	12,167	9,125	7,300	6,083
(3) R_d =	90	90	90	90	90
(4) k_1 =	1.02	1.02	1.02	1.02	1.02
(5) $\frac{v}{V}$ = lower limit line, Fig. 4	.512	.6	.672	.732	.79
(6) x for $\frac{v}{V}$	1.01	1.04	1.08	1.12	1.19
(7) $R = R_d \div \left\{ k_1^3 \times \left(\frac{v}{V}\right)^x \right\}$	166.7	144.3	130.3	120.3	112.3
(8) T.S. = $\pi \times R \times 16'.5$	8,644	7,478	6,753	6,235	5,820
(9) P.A. ÷ D.A. for T.S. (Fig. 2)	.403	.34	.298	.273	.262
(10) P.C. for P.A. ÷ D.A.	.615	.656	.678	.686	.69
(11) I.H.P. = E.H.P. ÷ P.C.	29,674	18,547	13,459	10,640	8,816
(12) K =	1.22	1.22	1.22	1.22	1.22
(13) K × I.H.P. = (actual power) =	36,203	23,803	16,420	12,977	10,756
(14) z	.728	.5493	.4238	.3266	.2432
(15) K × $I.H.P._d$ =	6,772	6,720	6,188	6,118	6,144

Plotting the K × $I.H.P._d$ and $\frac{e.h.p.}{E.H.P.}$ on P.A. ÷ D.A. as abscissas, the minimum value of K × $I.H.P._d$ is found to be:

$\text{K} \times \text{I.H.P.}_d = 5{,}980. \quad \text{I.H.P.} = 12{,}550$

$\text{P.A.} \div \text{D.A.} = .285$

$$\frac{e.h.p.}{\text{E.H.P.}} = .455$$

Now proceed as follows:

(16) P.A. ÷ D.A.	.285	.285	.285	.285	.285
(17) v	14	14	14	14	14
(18) I.T._D for P.A. ÷ D.A. (Fig. 2)	3.46	3.46	3.46	3.46	3.46
(19) P.T._p (Ch. 5, "Screw Propellers")	8.27	8.27	8.27	8.27	8.27
(20) E.T._p for Slip B.C. (Ch. 5)	9.49	9.59	9.49	9.49	9.49
(21) 1 − S	.8714	.8714	.8714	.8714	.8714
(22) $\frac{v}{\text{V}}$ (Fig. 4)	.7	.75	.775	.8	.86
(23) $\text{V} = v \div \frac{v}{\text{V}}$	19.59	18.92	18.3	17.73	16.5
(24) $\text{P} \times \text{R} = (\text{V} \times 101.33) \div (1 - \text{S})$	2,278	2,171	2,101	2,035	1,893
(25) D = *$\sqrt{(291.8 \times k_1^6 \times \text{I.H.P}) \div (2\,\text{I.T.}_D \times \text{P} \times \text{R})}$	16′.81	16′.57	16′.84	17′.11	17′.74
(26) x for $\frac{v}{\text{V}}$ and $\frac{e.h.p.}{\text{E.H.P.}}$ (Fig. 4)	1.09	1.28	1.36	1.48	1.75
(27) T.S. for P.A. ÷ D.A. = .285	6,340	6,340	6,340	6,340	6,340
(28) $\text{R} = \text{T.S.} \div \pi \text{D}_d$	124.7	121.8	119.8	117.9	113.7
(29) $\text{P} = (\text{P} \times \text{R} \times \pi \text{D}_d) \div \text{T.S.}$	18′.26	17′.82	17′.53	17′.27	16′.64
(30) $k_1 =$	1.02	1.02	1.02	1.02	1.02
(31) $\text{R}_d = k_1^8\, \text{R} \times \left(\frac{v}{\text{V}}\right)^x$	91.81	89.4	89.9	89.94	92.7

Plotting the curves of pitch and revolutions on diameter as abscissas, the resulting propeller for 16.5 feet diameter, is found to be:

$$\text{D} = 16.5 \qquad \text{R} = 90$$
$$\text{P} = 18 \text{ feet} \qquad \text{K} \times \text{I.H.P.}_D = 5{,}980$$
$$\text{P.A.} \div \text{D.A.} = .285 \qquad v = 14 \text{ knots.}$$

Note: All the propellers embraced in the limits of the last computations are of equal propulsive efficiency.

RELATIVE EFFICIENCIES AT VARYING LOADS.

The following table gives the relative efficiencies obtained by any projected area ratio for all conditions of load from $\frac{e.h.p.}{\text{E.H.P.}} = .1$ to $\frac{e.h.p.}{\text{E.H.P.}} = 1.1$, when delivering the same value of $e.h.p.$

*This equation is in error; k_1^6 should not appear in the equation.

$\frac{e.h.p.}{E.H.P.}$	$\frac{I.H.P._d}{I.H.P.}$	$\frac{e.h.p.}{E.H.P.}$	$\frac{I.H.P._d}{I.H.P.}$
.1	.9402	.7	.9731
.2	.9354	.8	.9692
.3	.9410	.9	.9812
.4	.9422	1.0	1.0
.5	.9428	1.05	1.0
.6	.9507	1.10	1.006

where I.H.P. is the power required to deliver *e.h.p.* under Chart Condition, which is where *e.h.p.* = E.H.P., and $I.H.P._d$ is the power required to deliver *e.h.p.* where *e.h.p.* is less than the Chart load, the values *e.h.p.* in all cases being equal.

It is seen that there is very little change in efficiency in passing from .1 load to and including .5, the maximum efficiency for all projected area ratios being at .2 load. Both the .1 and .2 load points would, however, produce very large propellers, and it is preferable, when designing for maximum efficiency, to use the .3, .4 and .5 load conditions, passing to the .6 conditions should the propellers given by the lower points prove too large in diameter for the case under consideration.

This will be the method pursued in the next problem, and this second method, may be preferable to that given in the problem just discussed.

Problem in designing for two conditions, as in cases of submarines and tug boats, or of vessels where maximum efficiency is desired at some other than the maximum speed.

To illustrate this problem, assume the case of a submarine vessel, the contract for which makes the guarantee for the submerged speed of much more importance than the guarantee of surface speed, so that it becomes necessary to design for the maximum efficiency in the submerged condition. The statement is as follows:

Slip B. C. of vessel = .35.

Guaranteed submerged speed = 10½ knots.

Total effective horsepower for this condition = 330.

Guaranteed surface speed = 14 knots.

Total effective horsepower for this condition = 530.

Required the propeller necessary to deliver the maximum possible propulsive efficiency at the submerged condition, also the revolutions and shaft horsepower for this condition.

Required the necessary shaft horsepower and revolutions for a surface speed of 14 knots.

The vessel to be fitted with twin propellers.

e.h.p. on 1 propeller = 165 at 10½ knots.

Assume, to obtain load condition for max. efficiency:

e.h.p. ÷ E.H.P.	.3	.4	.5	.6
E.H.P. = *e.h.p.* ÷ $\frac{e.h.p.}{\text{E.H.P.}}$	550	413	330	275
Assume, P.A. ÷ D.A.	.3	.3	.3	.3
P.C. for P.A. ÷ D.A.	.677	.677	.677	.677
I.H.P. = E.H.P. ÷ P.C.	812	610	487	406
z	.5493	.4238	.3266	.2432
K	1	1	1	1
$\text{I.H.P.}_d = \text{I.H.P.} \div 10^z$	229	230	230	232

As the submerged load is approximately equal to .6 of the surface load, and as it is desirable to keep the diameter as low as possible, use the .5 condition first, as it will also give a slight gain at the surface.

I.H.P.	487.	487.	487.	487.
P.A. ÷ D.A.	.3	.3	.3	.3
I.T._D for P.A. ÷ D.A.	3.77	3.77	3.77	3.77
P.T._P	8.49	8.49	8.49	8.49
E.T._P	10.31	10.31	10.31	10.31
1 − S	.8235	.8235	.8235	.8235
v	10.5	10.5	10.5	10.5
$\frac{v}{V}$ (Fig. 4) for $\frac{e.h.p.}{\text{E.H.P.}} = .5$	.732	.78	.82	.855
V	14.35	13.46	12.81	12.28
P × R = (V × 101.33) ÷ (1 − S)	1,765	1,657	1,576	1,511
$k_1 = 1$	1	1	1	1
$D = \sqrt{(291.8 \times \text{I.H.P.}) \div (\text{I.T.}_D \times P \times R)} =$	4′.621	4′.770	4′.891	4′.995
T.S. for P.A. ÷ D.A.	6,670	6,670	6,670	6,670
P	3′.841	3′.640	3′.630	3′.555
R	459.3	444.8	434.1	425.1
x for $\frac{v}{V}$ and $\frac{e.h.p.}{\text{E.H.P.}} = .5 =$	1.125	1.3	1.5	1.92
$R_d = R \times \left(\frac{v}{V}\right)^x$	323.5	322.	322.3	314.7
$\text{S.H.P.}_d = 230 \times .92$	212.	212.	212.	212.

Should these revolutions be too high, in order to go to lower revolutions it would be necessary to try again, using a load factor $\frac{e.h.p.}{E.H.P.} = .1$ or $.2$, depending on the amount of reduction desired. This could still further be decreased by decreasing the value of P.A. ÷ D.A. to say .25.

Assuming, though, for sake of illustration, that the revolutions obtained in the foregoing work were satisfactory, to obtain the surface condition proceed thus, using the load condition and propeller where V is greater than 14, namely,

$$v = 14$$
$$V = 14.35$$
$$\frac{v}{V} = .975$$
$$\frac{e.h.p.}{E.H.P.} = \frac{530}{550} = .964$$
$$x \text{ for } \frac{v}{V} \text{ and } \frac{e.h.p.}{E.H.P.} = 2.15$$
$$z \text{ for } \frac{e.h.p.}{E.H.P.} = .964 = .022$$

$$S.H.P._d = (487 \div 10^z) \times .92 = 426$$

$$R_d = R \times \left(\frac{v}{V}\right)^x = 459.3 \times \overline{.975}^{2.15} = 435.$$

In using Fig. 4, it is necessary to remember the following:

1. With constant value of $\frac{e.h.p.}{E.H.P.}$ and of $\frac{P.A.}{D.A.}$, the diameter of the propeller will decrease, the pitch and revolution increase as $\frac{v}{V}$ decreases.

2. With constant value of P.A. ÷ D.A., the revolutions will decrease while the diameter and the pitch will increase as the value $\frac{e.h.p.}{E.H.P.}$ decreases.

3. With constant values of *e.h.p.* ÷ E.H.P. and of v ÷ V, the pitch and diameter of the resultant propellers will

decrease as the projected area ratio is increased, while the revolutions will rapidly increase.

CONCLUSION.

The foregoing work is submitted as at least a partial solution of the "Mystery of the Screw Propeller."

It shows—

1. That the efficiency of performance of the propeller is seriously affected by its position in relation to the hull of the vessel which it is driving, and by the fullness of the hull lines at the after body.

It shows—

2. That for any given propeller working under constant hull conditions of form, the effective horsepower for any given engine power remains constant, and this is independent of the speed of the vessel.

It shows—

3. That for any given propeller working under constant hull conditions of form, the revolutions necessary to deliver any given effective horsepower vary with the speed of ship, the engine power remaining constant.

It shows—

4. That model tank effective horsepower curves are correct, and that the appendage resistance varies according to the "laws of comparison."

It shows—

5. That trials of vessels over shallow water courses should be prohibited, as such courses change completely the character of the water flow to the propeller.

It shows—

6. That deep-water trials, by the close agreement of actual with estimated results, demonstrate the correctness of the above statements.

It shows—

7. The great accuracy of torsion meters and indicators used on official trials, and that the large estimate of from three to

five per cent. error usually credited to these instruments is incorrect, particularly at high powers.

It shows—

8. That results obtained from model-tank experiments with model propellers will be correct when the model is properly proportioned to the full-size propeller, *which does not mean that the pitch ratio of the propeller and of the model shall be the same,* and the Chart Condition of the model propeller, working with no hull, or rather behind a phantom ship, can be calculated.

It shows, finally—

9. *That where vessels are entered into competition based upon relative efficiencies of their propelling machinery, it is a positive error to base these efficiencies upon speed or revolutions, as both speed and revolutions for equal load upon the propellers, represented by equal effective horsepowers, vary with the loading of the hull, foulness of ships' bottoms, weather and sea, while the engine power for this effective horsepower remains constant, except in cases where propellers are foul or their blades distorted. It shows, therefore, that the true criterion for efficiency of propelling machinery should be indicated or shaft horsepower developed by that machinery, and that the speed of ship and revolutions of propellers should be neglected.*

DESCRIPTION OF MAIN PROPELLING MACHINERY FOR THE U. S. S. *MAUMEE*.

BY LIEUTENANT C. W. NIMITZ, U. S. N., MEMBER.

The U. S. S. *Maumee* is one of two twin-screw fuel ships authorized by the Naval Appropriation Act of August 22, 1912, "to cost, exclusive of armor and armament, not to exceed one million one hundred and forty thousand dollars each, and which shall be built in navy yards, one to be built in a navy yard on the Pacific coast." These two vessels, the *Maumee* and her sister ship, the *Kanawha*, were built in the Mare Island Navy Yard to carry fuel and minor supplies for the fleet.

The *Kanawha*, now in commission with a merchant complement, is propelled by two triple-expansion steam engines of 2,600 I.H.P. each, while the *Maumee* is to have for propelling machinery two Diesel engines of 2,500 B.H.P. each. At the time these vessels were authorized there were already afloat a number of motor ships whose performance warranted the adoption in our service of Diesel motors of the heavy, slow-speed type for our auxiliary vessels.

The sum of money available for the construction of the *Maumee* was based on steam-engine propulsion and was not sufficient to cover the entire cost of experimental and construction work of a pair of large Diesel engines.

In order to introduce this type of motive power into our service, a Naval Appropriation Act of March 4, 1913, provided "that the un-obligated balances under the appropriation 'Steam Machinery' for the fiscal years ending June thirtieth, nineteen hundred and twelve, and June thirtieth, nineteen hundred and thirteen, not exceeding $250,000, are hereby reappropriated and made available for the development of a type of heavy-oil engine suitable for use in one of the fuel ships

authorized by the Act approved August twenty-second, nineteen hundred and twelve, and the expenditure thus incurred shall not be a charge against the limit of cost of such vessel."

It is the present intention of the Navy Department to man the *Maumee* with officers of the Navy and an enlisted crew.

The two ships are similar except as to their propelling machinery, which, however, will be kept within the same machinery spaces on both vessels, that is, between Frames 136 and 166. Externally their appearance will be the same, except that the smokepipe of the *Maumee* will come out of the after end of the machinery space while that on the *Kanawha* will come out of the forward part of the machinery space. The *Kanawha* has four Babcock & Wilcox boilers forward of the main engines, while the *Maumee* will have two boilers of the same make but smaller in size just abaft the main engines.

These two vessels will afford an excellent opportunity of comparing the two types of machinery, as they are not only sister ships, but they will be employed on identically the same duty and will probably burn the same grade of fuel. It would be premature at this writing to attempt any comparison between the two vessels, except that the *Maumee's* propelling plant will be more costly and considerably heavier than that of the *Kanawha*. A comparison of weights, however, should also include the fuel to cruise considerable distances, in which latter case the question of weight will be in favor of the *Maumee's* plant, owing to the unquestioned higher fuel economy of the Diesel motor over that of the steam engine.

The principal characteristics of these two vessels are given as follows:

Length between perpendiculars, feet	455
Beam, feet	56
Draught (loaded), feet and inches	26–4
Tonnage (loaded), about, tons	15,000
Speed (loaded), knots	14.5
Cargo capacity, fuel oil, tons	10,000

The main propelling machinery of the *Maumee* is being built at the Navy Yard, New York, and consists of two single-acting, two-cycle, six-cylinder, Diesel engines of the slow-speed, cross-head, heavy-duty type. Each engine will develop its full power of 2,500 B.H.P. at 130 r.p.m.

In order to gain a maximum of Diesel engine experience in a minimum of time, the Bureau of Steam Engineering decided to obtain the manufacturing rights and a set of working plans from some company already engaged in that line of work, and to build the engines in one of the navy yards. The Navy Department, therefore, obtained bids from various shipbuilding and manufacturing companies, both at home and abroad, and finally contracted with the Electric Boat Company, of Groton, Connecticut, American licensees of the Maschinenfabrik-Augsburg-Nürnberg, or M. A. N. Co. of Nuremberg, Germany.

The Electric Boat Company, through a subsidiary company, the New London Ship and Engine Co., manufactures two-cycle, single-acting, Diesel engines of the M. A. N. type for submarines and other marine work. This company also manufactures a four-cycle Diesel engine for marine use, but up to date has not constructed engines of the heavy-duty, slow-speed type used in ocean-going vessels.

The contractor obtained the original design and drawings from the home company, translated all figures from the metric to the English system, and supplied the Government with a set of retraced plans, whereon only slight departures from the original design were made. The Navy Yard, New York, checked the translated drawings by comparison with the original German drawings and, after approval by the Bureau of Steam Engineering, built the engines, following the drawings as closely as conditions would permit. It will not be out of place here to observe that it would probably have been cheaper to build the engines using the original metric figures and scales. All the translated figures contained decimals to the thousandth part of an inch, which made the checking ex-

tremely tedious and the work in the shop, where English scales were used, slower than it would have been otherwise. The decimals made it difficult for the man at the machine tool to distinguish between important and unimportant dimensions, so that the result showed considerable time spent in making all dimensions correct.

All parts of the engines were made in the navy yard, except the very few steel castings required, and the heavy connecting rod, piston rod and thrust and line-shaft forgings. These forgings were purchased rough turned and finished in the yard. The crank shaft was purchased finished from the Erie Forge Company of Erie, Pa.

Each main engine, turning outboard going ahead, drives the following attached auxiliaries:

Three high-pressure air compressors for fuel injection and for charging air-starting bottles.

Three scavenger compressors for cylinder scavenging.

Two salt-water pumps for cooling all water jacketed parts of engine except pistons.

One fresh-water pump for piston cooling.

One lubricating-oil pump for thrust block, main, crank pin, and crosshead bearing lubrication.

Two general service water pumps for fire, bilge or sanitary purposes.

Twelve mechanical lubricators for cylinders and for minor bearing lubrication.

One fuel-oil supply pump for pumping fuel from ship's bunker tank to engine supply tanks.

Six fuel-oil measuring pumps for supplying fuel to cylinders from engine-room tanks.

One speed governor.

One tachometer.

One revolution counter.

Referring now to the general arrangement plans of the engines, it will be seen that the scavenging pumps are mounted on the outboard columns of the even-numbered cylinders from

forward, and that they are driven by means of links and beams from the main crossheads of the engine. Each scavenger compressor is double acting and draws the air for both sides of the pistons through the common opening on the outboard side of the scavenger-cylinder casting (see general plan, "section through scavenger pump,") and then through twelve flat-disc suction valves, six for the top and six for the bottom of the compressor. The air, after compression to about eight pounds pressure, passes, via twelve discharge valves similar to the suction valves, through two coolers to the scavenger receiver, from which its admission to the cylinders is controlled by the cam shaft. The suction and discharge valves are assembled in six units, each unit consisting of two suction and two discharge valves mounted on one stem. By removing the valve bonnets the valve units are readily accessible and easily removed. The scavenger cylinder is lubricated at about its middle by four supply pipes, equally spaced, from the mechanical lubricators.

Directly under each scavenger compressor and driven by the same crosshead that drives the scavenger, are two pumps, which are used as follows:

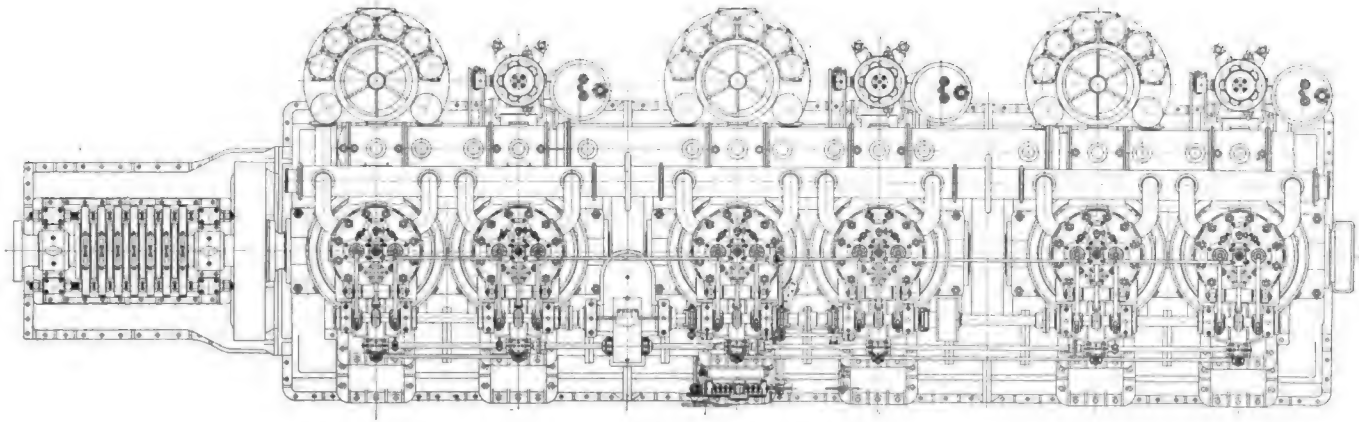
Under No. 1 scavenger compressor, forward pump for fresh water for cooling pistons; after pump for lubrication of main, crank pin, crosshead, and thrust block bearings.

Under No. 2 scavenger compressor, forward pump for salt water for cooling all parts of engine, except pistons; after pump, same as for forward pump.

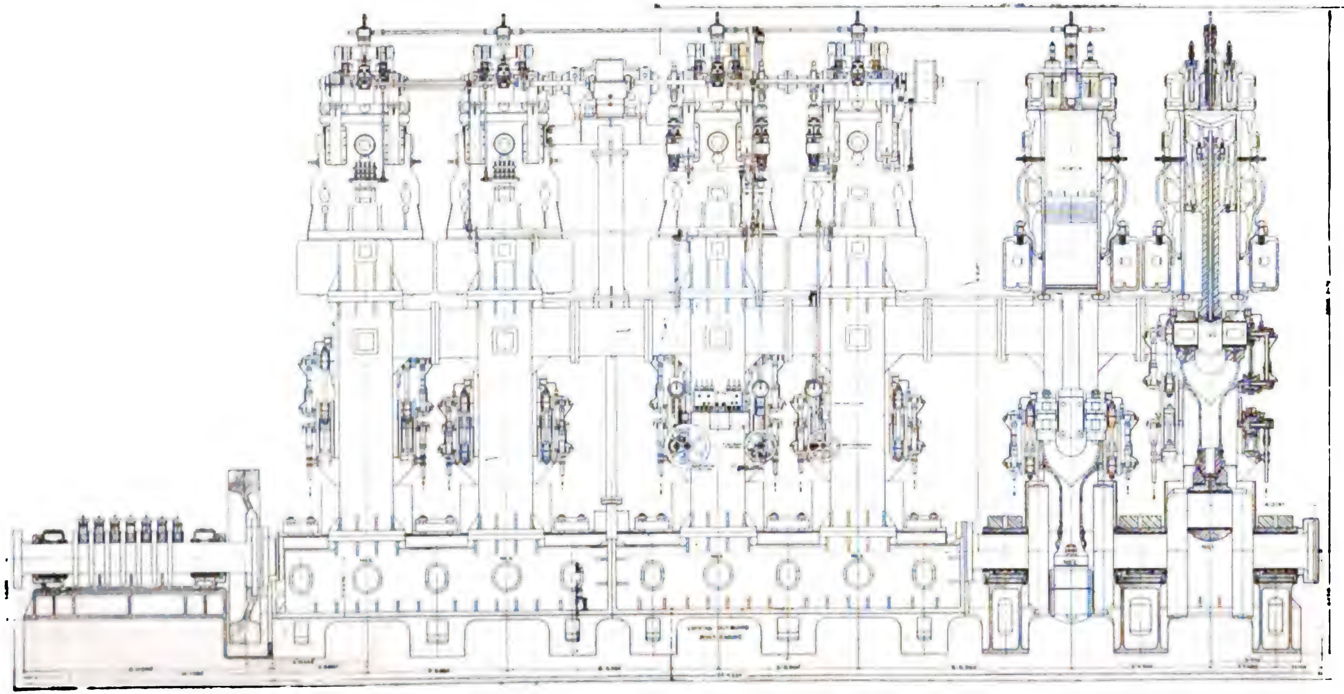
Under No. 3 scavenger compressor, forward pump for bilge and sanitary system; after pump for bilge and sanitary system.

All of the pumps under the scavenger compressors are of the same size and have their valve chambers attached to their forward and after sides respectively.

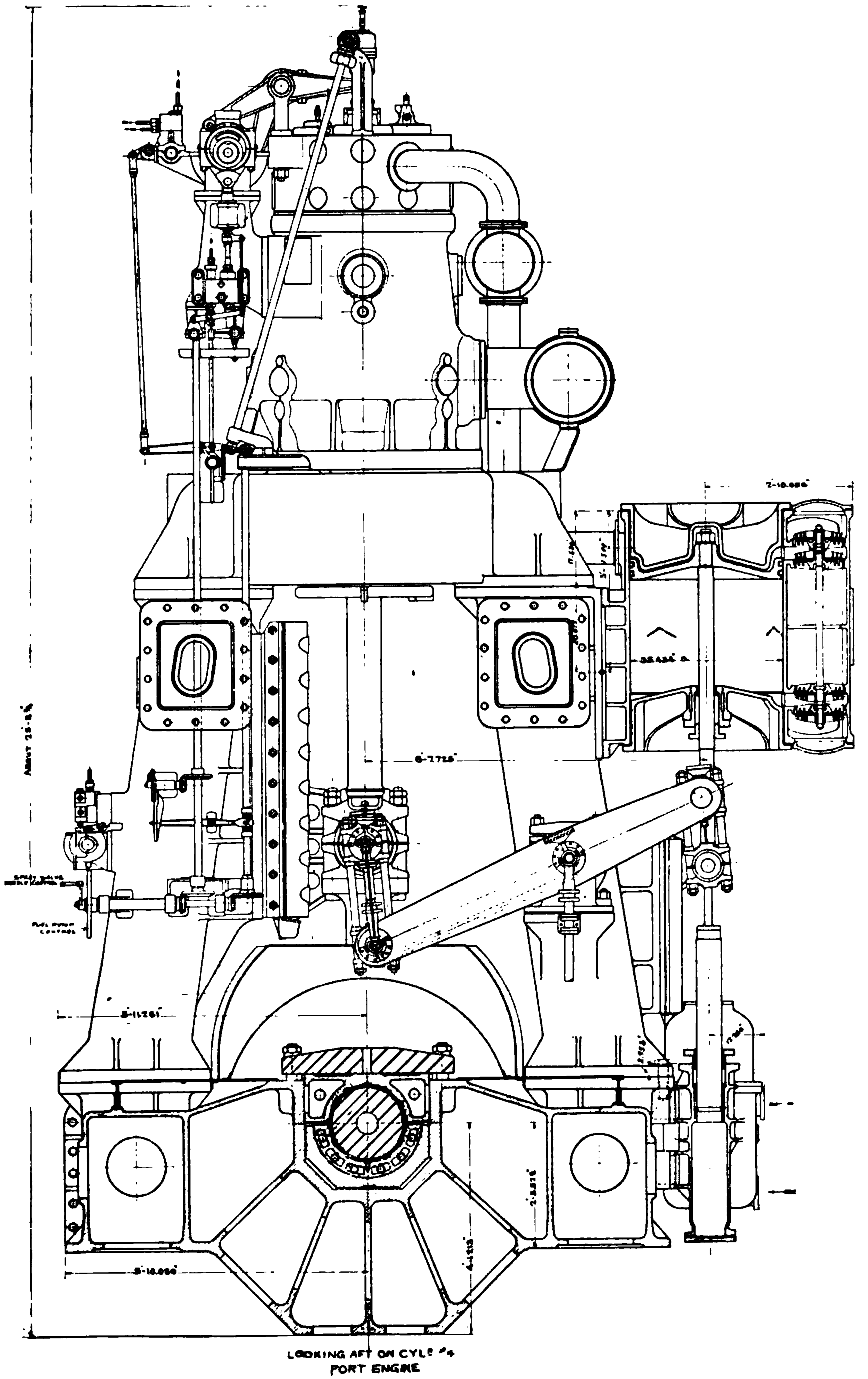
The high-pressure air compressors are mounted on the outboard columns of the odd-numbered cylinders and are driven off the main crossheads by links and beams. As in the case of the scavenger compressors, the beam at its outboard end



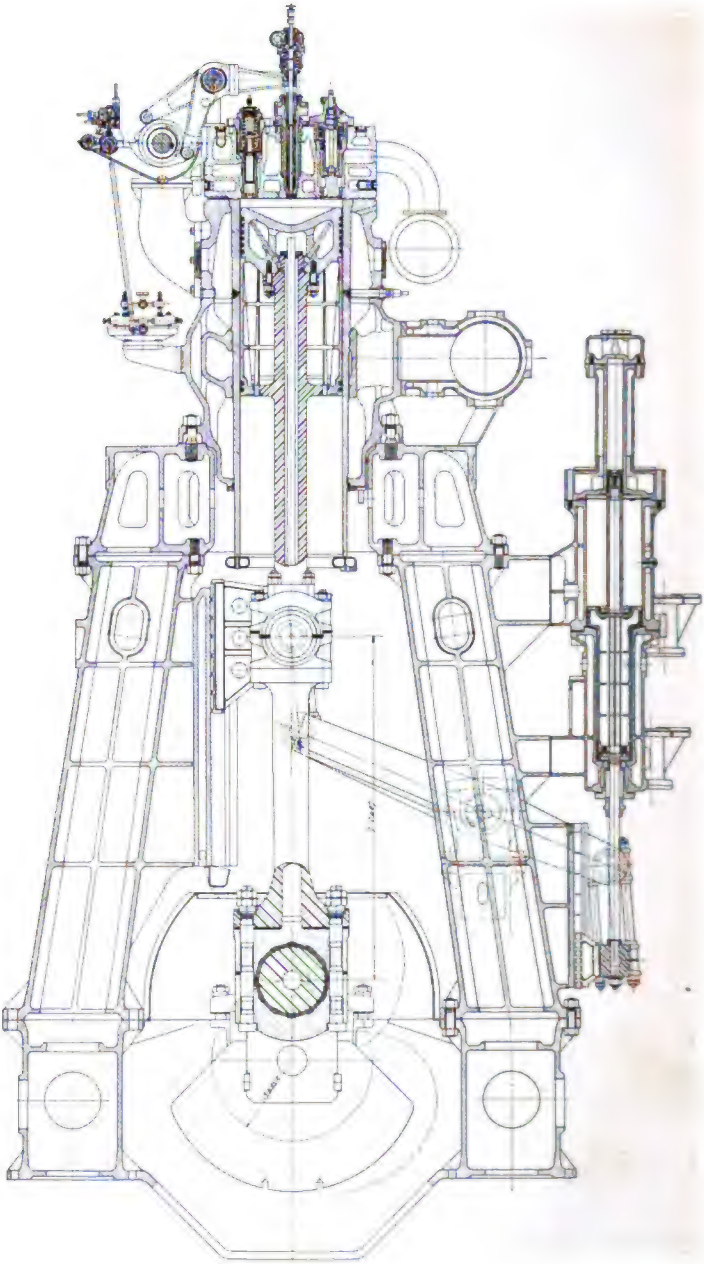
General Arrangement Plan.



General Arrangement.



SECTION THROUGH SCAVENGER PUMP.



SECTION THROUGH AIR COMPRESSOR
PORT ENGINE LOOKING AFT

Sectional Arrangement Through Working Cylinders.

drives a smaller crosshead traveling in a guide back of the column. Each compressor is of the three-stage, tandem-piston type. The low-pressure stage is double-acting, while the middle and high stages are single-acting. Each compressor has on its forward side an intercooler, which contains the cooling coils for all three stages. The piston and discharge valves of the low stage are of the flat-disc type, while those of the higher stages are of the poppet type. The three compressors deliver to a common pipe which conveys the air through separators to the spray-air bottle, from whence it leads to the fuel-valve bodies in the cylinder heads. The spray-air bottle has on it an overflow valve, whereby air in excess of the necessary spray air is passed into the bottles for storing the starting air. Two of the attached compressors are sufficient to supply injection air at full power while the third compressor acts as a reserve. The reserve compressor runs idle when the engine is running, although it can be easily disconnected from its crosshead. The low-pressure suction valves of all the high-pressure compressors are controlled from the handling platform of the engine as well as at each compressor. Any desired pressure up to the maximum can be maintained at will.

A small fuel-oil supply pump is attached to the after side of No. 1 outboard column and is driven by the beam. This pump maintains the fuel supply from the ship's bunkers to the engine-room supply tanks. The fuel-oil measuring or cylinder-supply pumps are six in number, one for each cylinder. Two pumps are contained in one body and all the pumps are driven by eccentrics from the main cam shaft. Fuel pumps for cylinders 1 and 2 are located on the after side of after cam-shaft bracket on No. 3 cylinder. These two pumps are in one heavy forging, both pistons being connected to the same crosshead. Fuel pumps for the remainder of the cylinders are similarly driven and are located as follows:

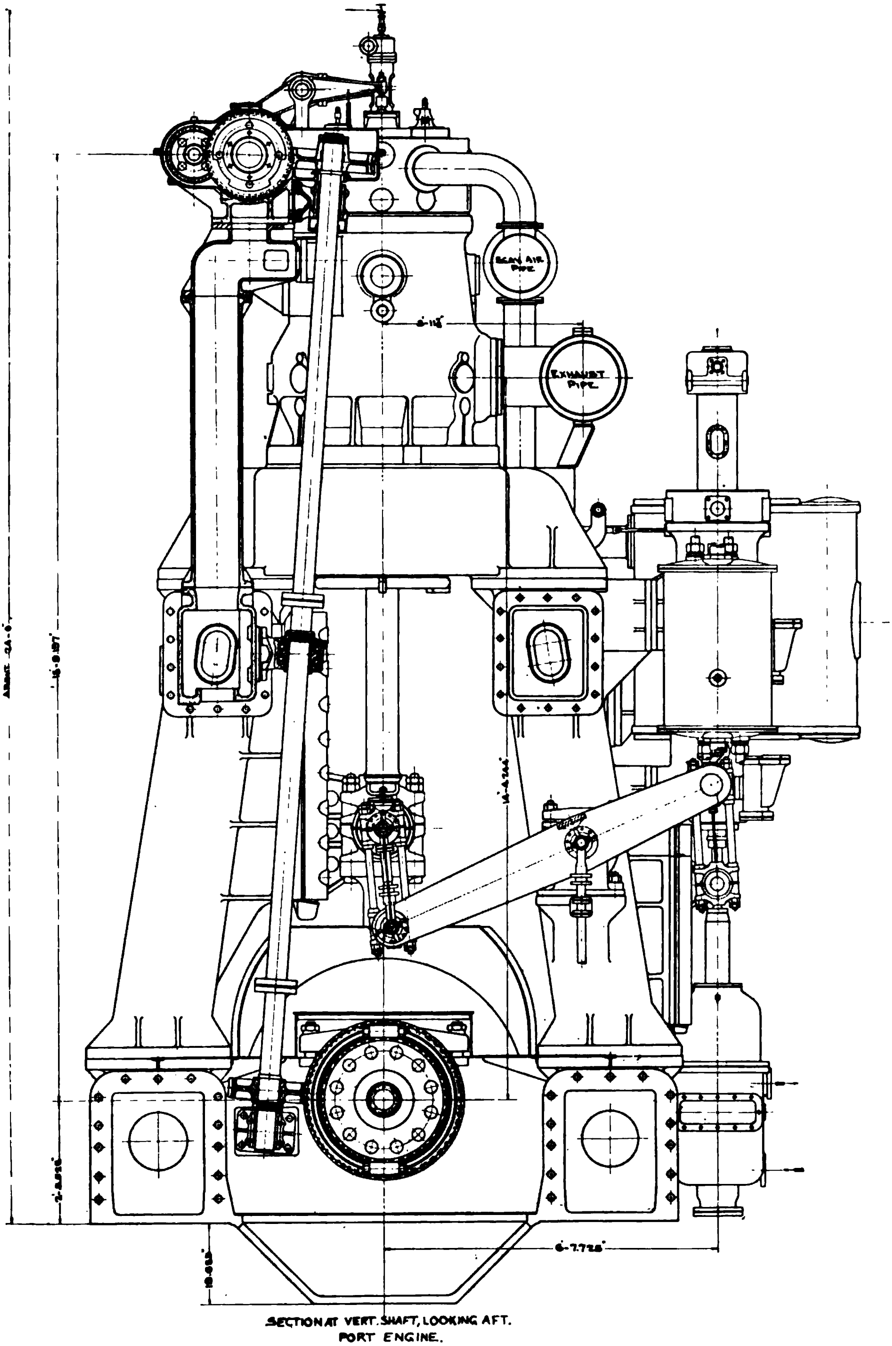
For cylinders 3 and 4, on forward side of forward cam-shaft bracket on No. 4 cylinder.

For cylinders 5 and 6, on after side of after cam-shaft bracket on No. 4 cylinder.

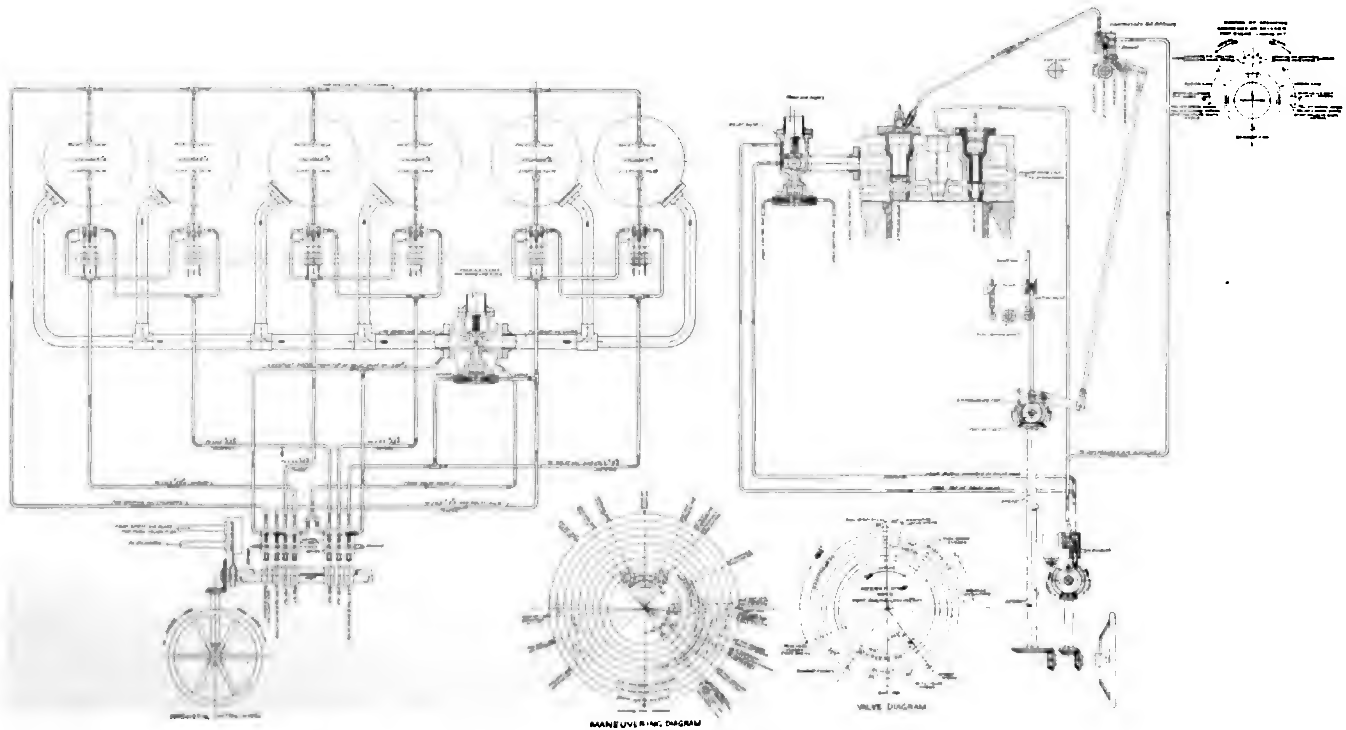
Each fuel pump has a mechanically-operated suction valve and two discharge valves in series. The pump delivers through a fuel-check valve held shut by spring and spray air pressure to the fuel-spray valve body in the center of the cylinder head. The same crosshead that drives the fuel-pump plunger also operates the lever that opens the suction valve. The inboard end of this lever acts as the fulcrum, and can be lowered or raised at will from the operator's platform, thus varying the period of opening of the suction valve. The speed and power of the engine are controlled by varying the period of opening of the suction valve, and hence the quantity of fuel pumped to each cylinder.

Before proceeding further it is proper to describe the cycle of events in the working cylinders of the engine on starting and in operation. Compressed air at about 650 pounds per square inch from the air-starting bottles is led to those cylinders whose cranks are in the proper position for running in the desired direction. After the engine starts to turn, starting air is admitted to each cylinder from 10 degrees past top center to 85 degrees past top center until the engine has attained sufficient speed for fuel admission. This speed is usually obtained in a very few revolutions. The operation of starting is quickly and easily accomplished in less than ten seconds, as will be explained later, the entire control being centered in one operating or "maneuvering wheel."

With the engine running normally, fuel is admitted to the cylinder once every revolution from 2½ degrees before top center to 37½ degrees after top center. Prior to the fuel admission clean air from the scavenger receiver has been compressed to about 450 pounds per square inch, at which pressure its temperature is about 900 degrees F. The fuel-spray valve is opened by the cam shaft through a lever, admitting the fuel to the cylinder, when the heat of the compression is greatest. The fuel enters in the form of a finely

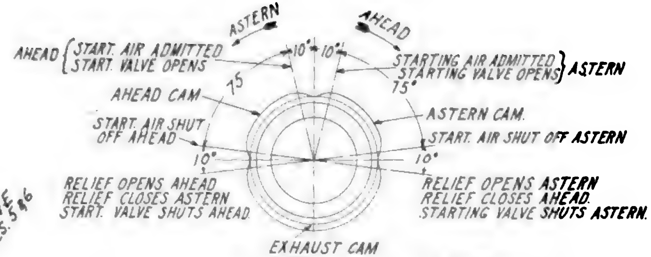


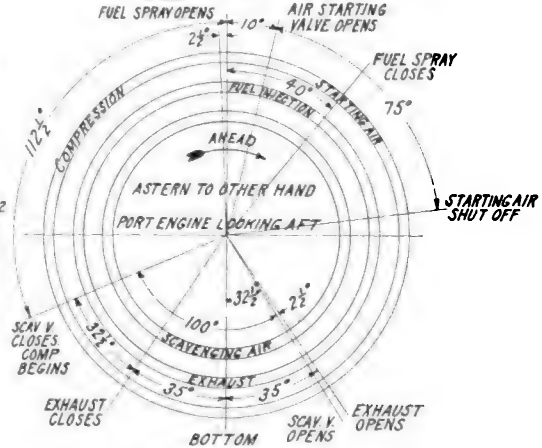
SECTIONAL ARRANGEMENT.



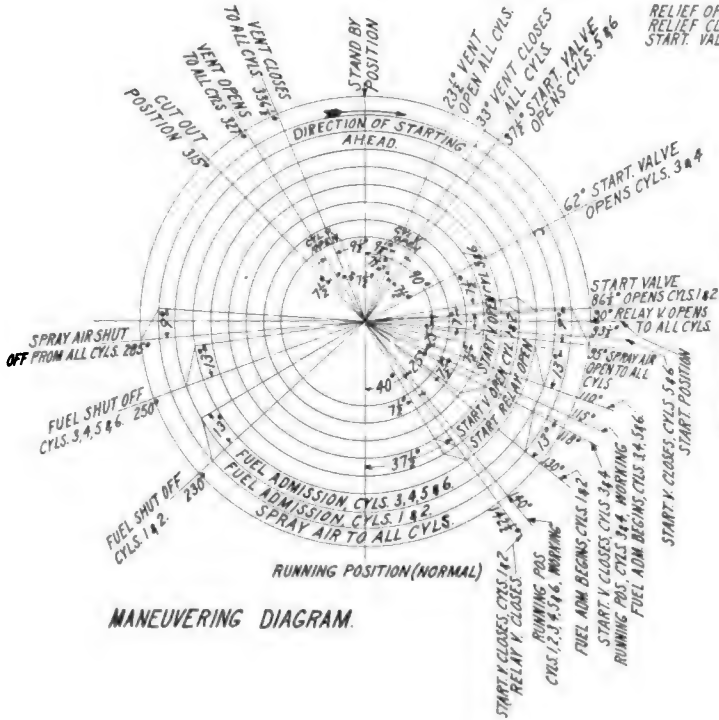
AIR STARTING DIAGRAM.

DIAGRAM OF OPERATION
COMPRESSED AIR DIFFUSER
PORT ENGINE LOOKING AFT.





VALVE DIAGRAM



MANEUVERING DIAGRAM.

divided spray, or mist, and begins to burn at once, the spray continuing until the fuel valve closes at 37½ degrees past top center. Expansion now occurs until 35 degrees before bottom center, when the piston uncovers a belt of exhaust ports in the cylinder. Two and one-half degrees after the exhaust starts to open, two scavenger valves in the cylinder head are opened by the cam shaft, admitting clean air at about seven or eight pounds pressure per square inch from the scavenger receiver. This clean air blows the burned gases out through the exhaust ports and fills the cylinder with clean air for another cycle of events. The exhaust ports are again covered by the piston at 35 degrees past bottom center and compression occurs. The scavenger valves remain open until 32½ degrees after exhaust ports are closed by the piston. Compression now occurs until 2½ degrees before top center, when the fuel valve opens.

The operation of reversal consists merely in cutting off all fuel from the engine, which causes it to stop very quickly, and then to start in reverse direction by compressed air as described above.

For the purposes of this description the engine will be divided as follows:

The main engine, consisting of six working cylinders;
The high-pressure air system;
The scavenging air system;
The salt-water cooling system;
The fresh-water cooling system;
The low-pressure lubrication system;
The high-pressure lubrication system;
The fuel system;
The maneuvering and control system.

THE MAIN ENGINE.

The main-engine bed plate has open crank pits and is of the type ordinarily used on marine reciprocating engines. There are three cast-iron sections bolted together, each section being

cored and ribbed for lightness and extreme stiffness. The crank pits being open, provision will be made on the ship for an oil basin underneath the engine. Each section has pads on its top side for bolting four uprights or columns, and on its outboard after side for securing the attached pumps under the scavenger cylinders.

There are three main bearings in each section of bed plate, the middle bearing being about 3½ inches longer than the end bearings, which are about 16½ inches long. Each main bearing consists of a flat-bottomed cast-iron seat, supported in the bedplate saddle; a lower main bearing brass cored for water circulation and capable of being rolled out of the saddle without removal of crank shaft; a flat topped upper-bearing brass which is cored out for stiffness and lightness; and a forged-steel binding cap. The bearing brasses are lined with a white metal of the following composition:

Tin	80 per cent.,
Antimony	15 per cent.,
Copper	5 per cent.,

this being somewhat harder than Navy-standard bearing metal. Four studs are provided for securing the forged-steel bearing cap. The flat-bottomed cast-iron bearing seat is capable of vertical adjustment to take up wear in main bearings, and the two bearing brasses are lipped at each end to prevent fore-and-aft movement.

The thrust bedplate is of cast iron, cored and ribbed for stiffness and bolted to the after section of the main-engine bedplate. It has a well at its forward end for the flywheel and engine-turning gear, the latter being driven by an electric motor just outboard the flywheel. The thrust block is of cast iron and of the ordinary marine type, and is secured to the table on the after end of the thrust bedplate. The block is bolted to the bedplate by clearance bolts and is capable of fore-and-aft and vertical adjustment. The block is prevented from fore-and-aft movement by two tapered keys fitted at

each end between the block and the thrust-bearing base. The thrust block carries seven cast-iron thrust collars of the horseshoe type, cored for water circulation and lined on the bearing faces with the special white metal mentioned above. The collars are capable of individual fore-and-aft and vertical adjustment on the thrust block, and are secured from fore-and-aft movement by a heavy, threaded rod on each side of the block. The thrust shaft, which is 14.567 inches in diameter, is supported in bearings at each end of the thrust block.

The main crank shaft, in three interchangeable sections, is 16.535 inches in diameter and is made of special forgings of the following characteristics:

Tensile strength, 71,000 to 78,000 pounds per square inch;
Elongation, 18 per cent. to 20 per cent.
Bending-test bar to be bent around an angle of 180 degrees cold.

The sections are hollow bored and drilled for the forced-lubrication system, the oil entering the main bearing journals from the lower main bearings. The sections are so bolted together that the sequence of the cranks turning outboard is 6, 1, 4, 5, 2 and 3. Counter weights are bolted and keyed to each crank shaft opposite the crank pin, and a flywheel, used also as part of the turning gear, is secured to the after end of the after section. The flywheel is made in halves, keyed and bolted together, in order to facilitate handling in the ship if it ever becomes necessary to remove it.

The main driving gears for the vertical shaft consist in a cast-steel helical gear made in halves and bolted around the coupling between the middle and after sections of the crank shaft, and a phosphor-bronze gear keyed to the bottom of the vertical shaft. The space between the middle and after sections of bedplate is made into an oiltight well for the gears to run in, the lower end of the vertical shaft being supported on the after side of the well. The lower bearing or support for the vertical shaft is itself secured in an eccentric bearing

set into the after bedplate, the object being to provide a means of taking up the wear between the steel and bronze gears.

The vertical shaft is on the inboard side of the engine and is in three sections, having keyed on its upper end a phosphor-bronze helical gear which drives the cast-steel gear on the cam shaft. The middle section of the vertical shaft has just below the upper coupling thrust collars which support the entire weight of the vertical shaft in a ball thrust bearing, which is in turn carried in a swivel bearing bracket bolted to the engine framing. The vertical shaft is again supported in its upper section by a swivel bearing bracket bolted to the steel casting forming a casing around the cam shaft and vertical shaft gears. The swivel bearing brackets are fitted with shims to provide for easy alignment.

The engine columns are made of cast iron, cored and ribbed for lightness and stiffness and are of the box pattern. The inboard columns carry the main crosshead guides, while the outboard columns carry the compressors for fuel injection and cylinder scavenging and the small crosshead guides for the auxiliaries. Outboard column No. 1 carries on its after side the fuel-supply pump driven off the crosshead of No. 1 cylinder and supplying fuel from the ship's bunker tanks to the fuel-supply tanks in the engine room. The outboard columns also carry on their forward and after sides the bearings for the beams. Inboard columns No. 3 and No. 4 carry on the lower platform all of the operating gear for handling the engine. Between columns 2 and 3, and 4 and 5, both inboard and outboard, are fitted cast-iron distance pieces to tie the columns in a fore-and-aft direction, forming a very rigid engine structure.

The cylinder bases are heavy iron castings, cored and ribbed, that tie the columns together athwartships and support the main working cylinders, these latter being secured by nuts and studs to the cylinder bases. The cylinder base has a hole in its center to allow the cylinder liner to extend below it.

The main crosshead slipper is of the usual marine type and is of cast steel, lined with hard white bearing metal.

The connecting rod and main crosshead, as well as the crank-pin and crosshead bearing brasses, are of the usual marine type, except that the bearings are lined with the special bearing composition and that the crossheads are bored out for the fresh-water passages of the piston-cooling system. The main crossheads have extensions fore-and-aft for driving the auxiliaries back of the engines by means of links and beams.

The piston rod is of forged steel and is hollow bored for the passage of the fresh water to and from the working piston. A nickel-steel pipe in the center of the piston rod conveys the water to the working piston while its return is effected by the concentric passage around the nickel-steel pipe. The upper end of the piston rod terminates in a heavy flange to which the working piston is securely fastened by means of nuts and studs. The lower end of the rod terminates in a square bolting flange for securing to the crosshead. A large round flange at about its middle serves to secure the lower part of the piston, which acts as a guide for the working piston.

The piston is divided into two parts, the working piston which consists of a special iron casting cored for water circulation and ribbed for strength; and a lower iron casting which is bolted to the piston rod. The two parts are not bolted to each other, although both are secured to the rod. The working piston is dished on top, and is machined with greater clearance at its top than at its bottom. It carries six cast-iron snap piston rings in grooves, which vary in width from the top on to the bottom one. The upper rings are given more clearance than the lower ones, on account of the greater heat. The fresh water coming up from the rod enters the central compartment of the piston, passes out towards the sides through cored passages at the top of the piston, and finally reaches the concentric space in the piston rod, via four pipes set at angles of about 45 degrees, returning from the

highest points of the water space, thus insuring a flow of water along the hottest parts of the piston. This method of conveying the cooling medium to and from the piston has the advantage of simplicity, but it has also the disadvantage of heating the water entering the piston by that just leaving the piston, and vice versa; the water just leaving the piston will not give a true indication of the piston temperatures as it has been somewhat cooled by the entering water. Eight large plugged holes in the bottom of the working piston afford easy access to all parts of the water space for removal of scale.

The lower part of the main piston, as has been explained before, serves only as a guide for the working piston. It is a cylindrical casting bolted at its lower end to a round flange on the piston rod. Its upper end is loose and free to expand, and fits over the lower end of the working piston. The guide piston is machined straight, with no taper to provide for expansion. Two cast-iron snap piston rings at the bottom of the guide piston prevent the escape of gas into the engine room. At the bottom of the stroke the top of the working piston just clears the ring of exhaust ports in the cylinder liner. At the top of the stroke the lower end of the guide piston just covers the exhaust ports.

The main cylinder is made up in two parts, a cast-iron jacket carrying the exhaust belt at its middle and a plain cylindrical liner of special cast iron.

The cylinder jacket carries twelve studs in its top flange for securing the cylinder head and a bolting flange near its bottom for securing to the cylinder base. At its middle it carries an exhaust passage which connects the cylinder exhaust ports with the exhaust piping. The space between the cylinder jacket and the liner forms the water jacket through which salt cooling water passes. The liner, which is thicker in section at its upper end, is pressed into the jacket until a shoulder on the liner rests on a shoulder in the upper part of the jacket. The top of the liner is securely held in place by the cylinder head while the lower end is free to expand

through the stuffing box in the bottom of the jacket, which prevents salt-water leakage. The bottom of the liner carries a double ring to catch salt-water leakage outside the liner or dirty oil from the inside of the liner. The oil and water thus caught is led to the bilge, from whence it can be pumped overboard. The liner has rectangular exhaust ports machined in a belt near its middle for the escape of the exhaust gases. The jacket is provided with eight handholes set 45 degrees apart, four in the upper part and four in the lower part, for the cleaning of scale from the liner. The jacket and cylinder are pierced at eight points 45 degrees apart for lubrication, four openings being in the upper part and four in the lower part of the cylinder. The press fit between liner and jacket is entirely at the exhaust belt, where every effort must be made to prevent the leakage of water from the cooling space into the exhaust ports. The surface of the liner passing through the tight fits at the exhaust belt has several shallow grooves whose object is to collect any slight water leakage, and "rust up," forming a watertight joint between liner and jacket. The surface of the jacket at the tight fits has annular grooves, one above and one below the exhaust belt. These grooves are about ¼ inch deep and ½ inch wide, and they are connected to pet cocks on the outside of the cylinder jacket. These pet cocks; kept open, act as leak indicators, should the water get by the other preventive measures. In case of leaks through the indicators, the proper procedure would be to gun the annular passages full of red lead, which would probably stop all leakage. A drain is provided at the lowest part of the water jacket. The cylinder liners are tested to 1,000 pounds per square inch above the exhaust ports, before being pressed into the jackets. A groove in the top of the liner forms a seat for the tongue on the bottom of the cylinder head. On the outboard side of the jacket is bolted an exhaust header of cast iron, cored for water circulation. Between the exhaust headers, sections of built-up exhaust piping are fitted, each section consisting of an inner and outer lap-welded steel pipe,

riveted to cast-steel flanges. The space between the inner and outer pipes forms the water jacket. The forward end of the exhaust pipe is closed by a water jacketed blank flange, while the after end is connected to the piping leading to the muffler. The exhaust headers and sections of piping are plentifully supplied with hand holes for cleaning the water spaces. On the inboard side of the cylinder jacket is bolted a bracket supporting a mechanical lubricator of the Detroit type. Two cast-iron brackets are bolted on the inboard side of each jacket near the top to support the cam-shaft bearings.

The cylinder heads are bolted to the cylinder by means of twelve long studs, the joint between the head and liner being made tight by a thin copper gasket, laid in the groove in the cylinder liner. The head has five openings in its top to receive the valve cages. The center opening is for the fuel-spray valve. The two largest openings are on either side of the spray valve in a fore-and-aft line and are for the scavenger valves. The inboard opening is for the cylinder-relief valve, while the outboard opening is for the air-starting valve. The scavenging-air receiver, consisting of a large copper pipe, runs the full length of the engine just outboard of the cylinders and near the heads. The receiver is supported by two pipes leading down to each scavenger pump, and by two pipes leading up to each cylinder head at the side, one for each scavenger valve in the head. The starting-air line runs the length of the engine and has a connection to each cylinder head at the side for the starting valve. The fuel-spray valve and the two scavenger valves are the only ones operated by the cam shaft. The starting valve is pneumatically controlled by the maneuvering wheel, but it may also be opened by hand. The relief valve is set to blow at 710 pounds per square inch, but it is also pneumatically operated by the maneuvering wheel.

Three types of cylinder heads are being tried on the first engine to be constructed. One type is shown on the general arrangement drawings, and is a special iron casting in one piece. The head is divided into two compartments for cool-

ing. The water from the cylinder jacket is by-passed around the cylinder-head joint into the lower compartment of the head, through which it must all go before rising to the upper compartment. A cast-steel sleeve in the center of the head receives the spray-valve body. Another type of head being tried is of the same design as that just described, but it has a much thinner bottom and is made of manganese-brass. The third type of head consists of a cast-iron upper story with a forged-steel bottom, the forged-steel bottom being machined out for water circulation. All the heads are drilled to receive pressure indicators and all water spaces are easily accessible for cleaning.

The relief valve consists of a forged-steel poppet valve, opening upwards, either by excess pressure in the cylinder or by compressed air acting on the bottom of the plunger attached to the valve spindle. The valve is carried in a separate valve body of cast iron, and is held shut by a spring. The upper part of the valve body contains a cylinder in which works the plunger attached to the valve stem.

The starting valve consists of a forged-steel poppet valve carried in a separate valve body of cast iron, the upper part of the body acting as a cylinder in which operates a plunger attached to the starting-valve stem. The valve opens downward, due to the admission of compressed air into the valve body on top of the plunger on the stem. The starting valve can also be opened by hand. The valve is ordinarily held shut by a spring.

The scavenger valves consist of forged-steel poppet valves, opening downward, due to the operation of the cam shaft and the scavenger levers. The valves are ordinarily held shut by springs acting on the bottom of a plunger or guide on the valve stem. The scavenger valves are housed in cast-iron valve bodies.

The fuel-spray valve, located in the center of the head, consists of a cast-iron body housing a long forged-steel needle valve opening upwards. This valve is opened by the cam

shaft and it is ordinarily held shut by a heavy spring. The compressed air for fuel injection is connected to the valve body at the top and maintains a constant pressure in the body, there being a safety valve in the air line at each cylinder. The fuel-oil line from the cylinder measuring pump is also connected to the top of the valve body, but fuel flows into the valve body only on the down stroke of the pump plunger, and then only when the engine is in the "operating" condition. As the fuel must be delivered to the valve body against the compressed air for injection, a check valve is installed in the fuel line as close to the valve body as possible. As it is extremely important that this fuel line be free of air at all times, a by-pass valve is fitted in the fuel line close to the fuel check but between the pump and the check valve. This by-pass permits the fuel to flow into a sight cup at the operator's station, and an inspection will indicate at once whether or not the fuel check leaks or whether the line is clear of air. The continuous presence of bubbles in the fuel from the by-pass indicates leaky fuel check, which must be ground in before the engine will operate properly. A small hand pump in the fuel line to the engine can be used preparatory to starting for priming the fuel system so that the engine will start on fuel with as few revolutions as possible. Whenever the fuel-spray-valve is raised, the fuel and air in the valve body are forced to pass through a series of perforated discs and by a cone-shaped atomizing device resembling a conical reamer. The mixture of air and fuel, after passing by the valve seat of the spray valve, enters the cylinder through an orifice in a spray button on the bottom of the valve body. The opening into the cylinder is cone-shaped with a view to injecting the mechanically-mixed air and fuel to all parts of the combustion space.

The inboard side of the cylinder head supports two stanchions which carry the shaft for the three valve levers for each cylinder, the middle lever operating the fuel-spray valve, while the other two operate the scavenger valves. Along the tops of the spray valves runs a shaft, which is turned by hand

from the operator's station. This shaft, by means of a worm and worm wheel in the top of each fuel-valve body, runs a stop up and down to regulate the stroke of the fuel needle. The fuel-valve lever in raising the valve acts on the heavy spring which holds the valve shut. When the stop is run down the fuel lever, after raising the valve the desired amount, merely compresses the spring without further movement of the valve.

The cam shaft is on the inboard side of the engine and is in four sections. The first section (from forward) carries the cams for cylinders No. 1 and No. 2. The second section carries the governor, the cams for cylinder No. 3, and an eccentric for driving fuel pumps for cylinders No. 1 and No. 2. The third section carries the cams for cylinder No. 4 and the gear which transmits the motion of the vertical shaft to the cam shaft. This section also carries a thrust collar to prevent lateral movement of the shaft. Section four carries the cams for cylinders No. 5 and No. 6. The cam shaft is supported in bearings contained in cast-iron casings mounted on the right-angled brackets on the inboard sides of the cylinders, two brackets being under each casing. The gear casing around the cam-shaft gear is somewhat larger than the others and is supported on a vertical box column bolted to the engine framing. There are three cams for each cylinder, two for scavenger valves and one for the fuel-spray valve. The cam shaft turns at the same speed as the crank shaft, but in the opposite direction. As the engine is reversible, provision must be made for utilizing the same set of cams for both ahead and astern operation. The cam-shaft gear is not keyed to the cam shaft, but it carries the male part of a two-jaw clutch. The female part of this clutch is keyed to the cam shaft and the female jaws are 35 degrees wider than the male jaws. Assume that the engine is running ahead and it is desired to reverse. The engine is first brought to a stop by cutting off the fuel and spray air. Compressed air from the starting flasks is then admitted to those cylinders whose cranks are in the right po-

sition for starting astern. The crank shaft moves through 35 degrees before the cam shaft starts to turn, when the cams will be in the same positions for running astern as they were for running ahead. The advantage of this method lies in its simplicity and the necessity for only one cam for each valve for both ahead and astern. Some engines have two sets of cams, one set for ahead running and one set for astern running. In this latter case the levers must be raised off the cams while the cam shaft is moved along its length to engage the second set of cams. The scavenger cams are solid steel forgings, case hardened, while the fuel cam is provided with an adjustable hardened steel toe, which permits of slight variations in the timing of the spray valves. The cams operate the valve levers through hard composition rollers in the inboard ends of the levers.

Bolted to the side of the cam-shaft gear is a smaller gear, which drives the starting shaft, just inboard of the cam shaft. As the cam-shaft gear turns at the same speed as the crank shaft, but opposite in direction, so the starting shaft turns in the same direction, as the crankshaft, and *always* with it. This shaft is mounted in bearings held in the cam-shaft casings and carries three cams directly underneath a distribution block for each cylinder. One cam operates a poppet valve in the distribution block for ahead motion of engine; one cam operates for astern motion, and the middle cam serves to vent the air piping which opens the starting valve. The distribution blocks are composition blocks forming part of the air-starting system and serve to open the proper starting valve on the cylinders at the proper time. As the engine starts to turn, the cams on the starting shaft, through the distribution blocks, hold open each cylinder-air starting valve from 10 degrees past top center to 85 degrees past top center, until the distribution blocks are rendered inoperative by means of the maneuvering control wheel.

Just inboard from the starting shaft is another shaft, called the "disengaging" shaft. This shaft, in two sections, is

operated entirely by the maneuvering wheel and serves to put in operation the distribution blocks for starting in either direction Cams on the disengaging shaft operate on bell-crank levers under the distribution blocks. The outboard ends of the bell cranks contain rollers which transmit the motion of the *starting-shaft* cams to the poppet valves in the distribution blocks. A flat spring under each block serves to hold the inboard ends of the bell cranks against the cams on the disengaging shaft. When the engine is running normally the cams on the disengaging shaft hold the rollers clear of the starting-shaft cams. The cams on the starting shaft, one for ahead, one for astern and one for venting the system for each cylinder, are so set that at least one crank will always be in the proper position to start the engine in the desired direction the instant starting air flows to the starting valves. The operation of the maneuvering wheel permits air to flow only to that end of the distribution block which will start the engine in the desired direction, the other half of the block being held open to the atmosphere.

THE HIGH-PRESSURE AIR SYSTEM.

The high-pressure air system of one engine consists of the three attached air compressors, the spray flask of about five cubic feet capacity, the six starting-air flasks with capacity of about 180 cubic feet, air separators, piping, relief valves, etc. One auxiliary air compressor, independently driven by steam, with capacity equal to that of one of the attached air compressors, is also provided for charging spray and starting flasks when all air is gone. The auxiliary compressor is of the three-stage, tandem-piston, slow-speed, horizontal type as manufactured by the Ingersoll-Rand Company. The suctions of the low-stage cylinders of all compressors will be taken from the coolest parts of the engine room or from outside the engine room if necessary on account of the reduced efficiency caused by sucking in high-temperature air. As the low-pres-

sure stage of the attached compressors is double acting, both suction pipes lead to a common valve which controls, via the operator's platform, the admission of air to the compressors. These suction valves can also be regulated at the compressors, independently of the gear at the operator's platform, so that any one of the three attached compressors can be made the reserve compressor. The valves of the low and intermediate cylinders are carried in detachable valve cages, making them very accessible for overhaul or examination. The valves of the high-stage cylinder are in the cylinder head and are easily accessible. The compressed air is cooled by its coil in the intercooler after each stage and is finally delivered to the collecting pipe back of the engine through a non-return valve. The collecting pipe, having received the air from all the compressors, delivers to a separator, where oil and water are partially removed. From the separator the air is led to an air manifold, into which all attached compressors of both engines and the auxiliary compressor supply the air. From the manifold the air may be led entirely to the spray flask or to the starting bottles or partially to each, the spray air being kept at the desired pressure while the balance of the compressed air is stored in the starting bottles. From the manifold the air for fuel injection is led to the spray flask. The spray-flask head is provided with a spring-loaded overflow valve, which permits any excess over the desired spray pressure to flow to the starting bottles via the manifold. The spray flask is also provided with a drain for freeing the air of oil and water. From the spray flask the air for fuel injection is led to an automatic cut-out valve at the control station, the object of which is to have the air on the injection system only when the engine is in the running condition. During reversal the spray air is cut off the engine until the crank shaft has started to turn in the desired direction. From the automatic cut-out the air is led to a manifold pipe at the top of the engine, from which a short lead supplies each fuel-spray valve body. A relief valve set at 1,200 pounds per square inch and a check

valve are placed in the line to each fuel-valve body, as close to the valve body as possible.

THE SCAVENGING AIR SYSTEM.

The scavenging or low-pressure air system consists of the three attached double-acting scavenger compressors, with their piping, relief valves, etc. The air, drawn from the engine room or from outside, passes through the suction and discharge valves of the scavenger compressors into the coolers, two in each scavenger cylinder casting. Each cooler consists of a nest of straight tubes with cooling water inside the tubes and air outside. From the coolers the air is led to the scavenger receiver, a large copper pipe running the length of the engine. The receiver is made up in three sections, with an expansion joint between the sections. At each end of the receiver is a large relief valve set at 15 pounds per square inch. Each expansion joint is fitted with a drain for the removal of water or oil in the air. From the receiver two connections to each cylinder head convey the scavenging air to the scavenging valves.

THE SALT-WATER COOLING SYSTEM.

The salt-water cooling system consists of two attached plunger pumps, under the middle scavenger pump, and an independently driven steam-plunger pump, together with the necessary connections and piping on the engine. The independently-driven steam pump will also be connected up to run by compressed air until steam can be gotten up on a boiler.

Both attached pumps have a common suction, each pump being of sufficient capacity to supply the salt-water system at normal power. The salt water is discharged into a large main back of the engine, underneath the floor plates. From the main a branch leads upwards to the bottom of each intercooler for the high-pressure air compressors and to the bottom of each cooler in the scavenger pump castings, making a total of nine outlets from the main back of the engine. The

main continues around the forward end of the engine, where a branch leads upward and aft on the outboard sides of the main bearing caps. This branch supplies water to the jackets of the lower main bearings and to the thrust collars on the thrust block.

The supply main continues around to the inboard side of the engine under the floor plates, a branch being led to the bottom of each ahead crosshead guide. A collecting main runs around the engine at the heighth of the cylinder bases. On the inboard side of the engine it receives the return cooling water from the crosshead guides. At the after inboard side it receives the return cooling water from the main bearings and thrust block. On the outboard side of the engine it receives the cooling water from the scavenger coolers. Back of the engine all of the water in the collecting main enters the bottom of the main cylinder jackets, two branches leading to each jacket. The cooling water leaving the high-pressure intercoolers of each compressor is led to the lower end of the jacket of the middle-stage air-compressor cylinder. From here it is forced upward into the jacket of the low-stage cylinder through two ferrules set partly into each cylinder at the joint. From the low-stage jacket the water enters the high-stage jacket through two by-passes around the cylinder joint. From the high-stage jacket the water is forced into the high-stage cylinder head, through two by-passes around the joint between head and cylinder. From the head of each high-stage cylinder the water is led into the exhaust-pipe jacket. This return water was originally forced into the above-mentioned collecting main from whence it entered the main cylinder jacket, but the heating of the compressors due to poor circulation of the water forced the change described above. From the main cylinder jackets the water enters the cylinder head through a by-pass around the joint between the cylinder and the head. After circulating through the lower and upper compartments of the head, the water enters the exhaust-pipe water jacket. From the exhaust-pipe jacket the

water is finally discharged into an overboard-discharge main. A small pipe leading from the cooling outlet of each cylinder head to a pet cock discharging into a funnel at the operator's station, indicates whether or not the cooling system is functioning properly.

THE FRESH-WATER COOLING SYSTEM.

The fresh-water cooling system is supplied by the forward attached pump on the engine and can also be supplied by the independently-driven steam pump provided for emergency fresh-water circulation of both main engines. The fresh water is drawn from a double-bottom compartment where it is stored, and discharged into a large main back of the engine and below the floor plates. From the main a connection leads to the after beam bearing on each outboard column. At this point the water passes, via swivel joint, through the bearing to a pipe secured to the beam. This pipe conveys the water to the crosshead end of the beam, where it again passes to the after side of the beam, via swivel joint, through the lower bearing of the link. Another pipe with swivel joint conveys the water into the after end of the main crosshead, from which point the water reaches the working piston by means of a hole through the crosshead connecting to a nickel-steel pipe running up the center of the piston rod. The water is delivered into the center compartment of the working piston, from which it passes to the outer compartment through cored passages at the top. Four collecting pipes, reaching the highest parts of the outer cooling space, return the fresh water to the center of the piston rod into a space around the upward conveying pipe. From the piston rod the hot water reaches a discharge main back of the engine, via links and beams and the forward end of the crosshead, in a manner similar to that of entering. The discharge main returns this hot water to the double-bottom compartment from whence it came. The water will be cooled by the ship's bottom and by introducing cooling coils, if found necessary. A small copper pipe lead-

ing from the discharge side of each piston, is led to the inboard side of column No. 2, where it delivers through a small pet cock into a funnel. The operator on the handling platform is thus able to see at a glance whether or not the system is functioning properly. He can also *feel* the temperature of the cooling water. A thermometer is installed in the supply main near the pump discharge and additional thermometers are installed in each outlet from the pistons.

The emergency independently-driven steam pump in the ship for fresh-water cooling will also be connected up to the compressed-air system, so that the pump can be operated by air until steam can be raised on a boiler.

THE LOW-PRESSURE LUBRICATION SYSTEM.

The low-pressure lubrication system is supplied by the after plunger pump under the forward scavenger cylinder. The pump sucks from a well in the engine crankpit and discharges through a strainer, and a Schutte & Koerting film cooler to the main, which runs the length of the engine just inboard of the main bearing caps and underneath the floor plates. From the main a supply pipe leads to the bottom of each main bearing brass. The main also delivers to a manifold which supplies the thrust collars and bearings. The oil flows into a drilled passage in the lower main-bearing brass casting, from which it enters a fore-and-aft groove in the bearing surface through two supply holes. Each main journal has four lubrication grooves, 90 degrees apart, machined into its surface. Each groove connects to the center of the shaft, through a drilled hole. Two of the grooves on the journal collect the oil from the forward end of the groove in the lower bearing, while the other two collect the oil from the after end of the supply groove in the main bearing. From the center of the main journal the oil passes to the crank-pin bearings through passages in the webs, and drilled holes in the crank-pin journals. From the crank-pin bearings the oil passes through a hole in the top brass to a pipe leading to the lower bearing of

each crosshead brass. From the crosshead the oil drops back to the crank pit, or is caught at the ends of the bearings and returned to the crank pit by pipes. The oil from the thrust block is also drained back into the crank pit. The emergency-lubrication pump on the ship will be independently driven by air as well as by steam.

THE HIGH-PRESSURE LUBRICATION SYSTEM.

The high-pressure lubrication system is separate for each cylinder and is supplied by a 32-feed Detroit mechanical oiler mounted on the inboard side of each cylinder and driven by bevel gears from the starting shaft. Each oiler receives its supply from a connection to the low-pressure lubrication system. Each oiler has eight connections to each main cylinder, six connections to the main crosshead guide, three connections to the auxiliary crosshead guide, four connections to the scavenger cylinders on even numbered cylinders, and various connections to link and beam bearings, etc. The high-pressure compressors have special gravity-feed lubricators attached to each cylinder.

The cam-shaft bearings are lubricated by gravity-feed cups.

THE FUEL SYSTEM.

The fuel system is supplied by two gravity tanks for each engine, these tanks being kept full by the attached fuel pump driven off the after beam of No. 1 cylinder. This pump draws the fuel from the ship's bunkers and delivers through a strainer to the engine-room gravity tanks. From the gravity tanks the fuel flows through a filter to the suction side of each cylinder fuel pump. The speed and power control of the engine by means of the suction valves of these pumps has already been explained. The cylinder fuel pumps discharge to their spray-valve bodies through check valves close to the fuel-spray valves. A by-pass valve in the fuel line at the highest point just before it reaches the fuel check permits the

fuel line to rid itself of all contained air, and also serves to detect leaky fuel checks, as explained before. The fuel-supply tanks contain heating coils in case it becomes necessary to heat the fuel. An oil heater of the straight-tube type is also installed in the line between supply tanks and cylinder-fuel pumps, the circulating water from the cylinder heads being used to heat the oil. A small hand pump for priming the fuel system is also introduced in the supply line between the gravity tanks and the cylinder-fuel pumps. Before starting the engine, it is necessary to open all by-passes in the fuel lines to cylinder heads and, by means of the hand-priming pump, fill the fuel line so that one or two revolutions will force fuel into the spray valves.

MANEUVERING AND CONTROL SYSTEM.

On the operator's platform, on inboard columns 3 and 4, are located the devices for the power and speed control and the maneuvering of the engine. On the after side of column No. 4 is the maneuvering-control wheel, which controls the starting, stopping and reversal of the engine by means of compressed air.

This wheel also cuts out the fuel and spray air from the cylinders during the maneuver and until the engine is turning over in the desired direction. Above the maneuvering control is a dial on which a pointer indicates the running positions of the engine.

On the forward side of No. 4 column is the fuel-control wheel, which governs the quantity of fuel pumped into each cylinder. A pointer and dial above the fuel control indicate arbitrarily, in eight equal steps, the quantity of fuel pumped, from the minimum to the maximum.

Coming out from the shaft of the fuel-control wheel is the needle-stroke control, which varies the stroke of the fuel spray needle from the maximum to the minimum. On the after side of column No. 3 is a hand cut-out, by means of which the

engine can be instantly stopped. It operates to raise the suction valves of the fuel pumps, which renders them inoperative.

Below the hand cut-out is the control for the high-pressure air supply, which regulates the openings of the suctions of the low-stage cylinders of the air compressors. The quantity and pressure of the spray air is controlled by this control.

The future of the Diesel engine of moderate power and speed of revolution is assured on land and on slow-speed ocean-going freighters or tankers of moderate size. Ship owners are keenly alive to the advantages of the motor ship over the steam ship, this being evidenced by the many orders for motor ships being booked by the Burmeister and Wain Company of Copenhagen, Denmark. The war in Europe has left this company in the enviable position of being the only company with motor-ship experience whose shops are available for commercial work.

The revival of the shipping industry of the United States will undoubtedly bring orders for motor vessels to those shipyards in this country prepared to build the new types of engines.

PRACTICAL LUBRICATION.

By Lieutenant G. S. Bryan, U. S. Navy, Member.

The average engineer can hardly be expected to make an extensive study of lubrication and there is very little literature on the subject that is written in a form that is of practical use to him. Nearly all articles on the subject are either too theoretical and technical or else they are written in the interest of a certain brand of oil, and the facts correspondingly distorted or misrepresented to fit that particular oil. This article is intended to give briefly and in as practical a way as possible the general information that is needed by the engineer to assist him with lubrication problems.

WHAT IS A LUBRICANT?

We all know that two metal surfaces rubbing together give rise to considerable friction and to reduce this friction we use a lubricant between them. The theory is that the oil forms a thin film that entirely separates the two metal surfaces, and the friction generated under these conditions occurs in the body of the lubricant itself, and is due to the molecules of oil rolling or sliding on each other. We might consider, as an illustration, that the two bearing surfaces float on the film of oil.

The fundamental characteristics that a lubricant must have then are

1st. It must be capable of forming a thin film between the two surfaces.

2d. It must resist being squeezed out from between them.

3d. It must have low internal frictional qualities.

Any substance that satisfies these three requirements can

be classed as a lubricant. There are other qualities that a *good* lubricant should have, but the above three are the fundamental ones. About the only substances that possess these qualities to a sufficient degree to be of practical use are the so-called mineral, animal and vegetable oils.

Formerly the animal and vegetable oils were used almost exclusively for lubricating engines and machinery. Of late years, particularly since the advent of forced-feed lubricating systems, mineral oils have come into general use on account of their cheapness as well as other properties which make them more suited for general lubrication. In the naval service all oils used are pure mineral oils except that used for marine engines without forced lubrication, this oil containing a small percentage of blown rape-seed oil.

VISCOSITY INDICATES THE FRICTION.

The characteristic that indicates the internal frictional qualities of an oil is called the *viscosity*. It is measured by the time required for a certain amount of the oil to flow through a small standard orifice in an instrument called a viscosimeter. The viscosity can be judged roughly by noting the sluggishness with which an oil flows, and two samples can be compared in this respect by suddenly inverting sample bottles containing them and noting the comparative rapidity with which the entrapped air bubbles rise to the surface. In general the more viscous the oil the greater the friction generated and the greater the ability to withstand being squeezed out from the bearing.

The viscosity of an oil varies with the temperature, however, and the friction will also vary correspondingly. The higher the temperature the lower the viscosity, until at about 300 degrees F. nearly all oils have a viscosity only a little greater than that of water. The variation with the temperature is shown graphically by the curves in Figure 1, which are made for typical oils.

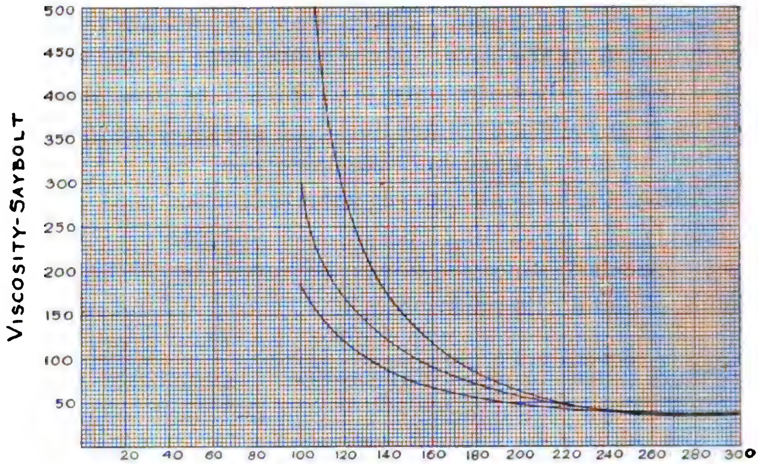


FIG. 1.—TEMPERATURE OF OIL—DEGREES FAHRENHEIT.

THE EFFECT OF TEMPERATURE.

The first effect of a high temperature in a bearing, then, is to thin the oil and thus decrease the friction. The effect of cooling the bearing is to increase the friction. There is no virtue in keeping a bearing at a very low temperature by the use of a considerable amount of cooling water, and there is no reason why a bearing should not be allowed to run at a high temperature as long as it is in good condition and is getting plenty of good clean oil.

The principal reason why a hot bearing is so feared is that it is generally an indication that something is wrong with the bearing or that it is not getting sufficient oil. Possibly this is a good reason for not allowing bearings to run at very high temperatures, but there is no excuse for going too far the other way and keeping them too cool. There is a turbo-generator in the power plant of the Naval Engineering Experiment Station at Annapolis, the bearings of which run regularly at temperatures between 190 F. and 210 F., and this has never given any trouble, although the oils used in it have ranged from the heaviest forced-lubrication oil to the lightest ice-machine oil in use in the service. In one case an

average temperature of 205 F. was maintained for 30 days. Cooling a bearing below a fair working heat by flooding it with oil results in a waste of oil, and keeping it cool by circulating a large amount of cooling water around it results in a waste of power. However, in case a bearing heats up due to the presence of dirt or foreign matter, an increased supply of oil may be necessary to wash this dirt out of the bearing.

THE EFFECT OF THE LOAD.

Theoretically, the total friction in a perfectly lubricated bearing is independent of the load that may be put on the bearing. This may appear surprising, but it nevertheless is true, and can be easily demonstrated. In practice, while we do not find this rule to hold exactly true, we do find that it holds approximately so. The coefficient of friction drops very appreciably with an increase in the bearing pressure, the coefficient of friction being defined as the ratio between the

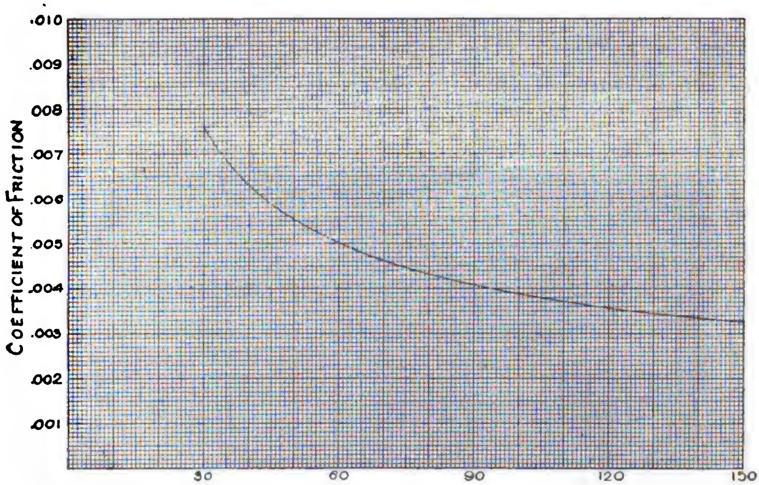


FIG. 2.—BEARING-CAP PRESSURE—POUNDS PER SQUARE INCH.

total friction and the total load. In Figure 2 a typical curve of the coefficient of friction is shown with points plotted for 30, 60, 90, 120 and 150 pounds per projected square-inch load

on the bearing, and this shows graphically the manner in which the above ratio varies with the load.

The principal danger of increased load on the bearing, then, is not that it directly increases the friction, but that it tends to squeeze out the oil from between the journal and bearing and thus destroy the lubrication. Generally speaking, the lighter (less viscous) the oil the easier it is to squeeze it out. It can be easily seen that the facility with which the oil is squeezed out naturally acts to increase the amount of oil used, since the oil forced out must be replaced by fresh oil. Forced lubrication is used mostly in the Navy now, and the loss would also be increased in this case by the more rapid circulation of oil through the system, since this would increase the leakage from imperfect joints, splashing of oil into the bilges by engine cranks, etc. This has caused many engineers to change to a heavier oil as a matter of oil economy.

THE EFFECT OF JOURNAL SPEED.

Increasing the rubbing speed of the journal greatly increases the friction. The curve in Fig. 3 illustrates graphically the amount of this increase. It will be noted that at

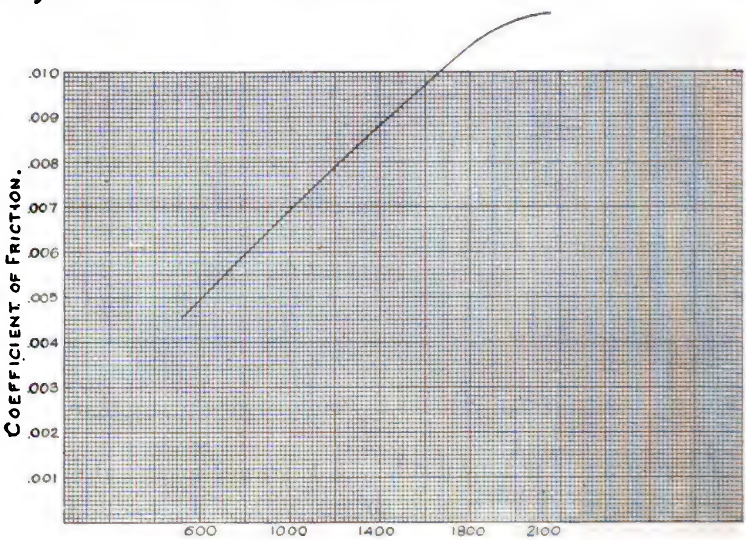


FIG. 3.—JOURNAL-RUBBING SPEED—FEET PER MINUTE.

rubbing speeds above 2,100 feet per minute the curve appears to flatten. This speed is exceptional, however, and at the speeds ordinarily encountered in the Service the friction increases greatly as the rubbing speed increases.

SELECTING A SUITABLE OIL.

In deciding whether to use a high (heavy), intermediate, or low (light) viscosity oil on a particular bearing we have three factors to consider—(1) temperature, (2) pressure, and (3) journal-rubbing speed; and we have just discussed what will be the effect of varying these factors.

The temperature that we should consider is the one we expect the bearing to have under practical working conditions and we should choose our viscosity accordingly. If we have two bearings that are identical, one working at 100 F., and the other at 150 F., we should have an oil of a certain viscosity at *100* F. for the first bearing and one of the same viscosity at *150* F. for the second.

With regard to pressure, the important thing is to get an oil that has body enough to prevent it from being squeezed out under the maximum load to which the bearing will be subjected. Then if you find that you are using too much oil you can generally reduce the amount by substituting a heavier oil. This will save oil but will increase the friction slightly. The oil finally selected must necessarily represent a compromise between the different factors.

In a high-speed bearing it is necessary to use a thin oil, on account of the great friction due to this high speed. Bearings of this type are generally designed so as not to have a very great bearing pressure, so that a light oil can be used without danger of it being squeezed out.

In turbine bearings the best conditions are obtained for the use of light oils. Here the speed is high and the bearing pressure is not only small, but is also steady and uniform in character. In a reciprocating engine the rubbing speed is much slower and the pressure on the bearings is intermittent

in character. The reciprocating action intensifies the bearing pressure and the squeezing out effect, but has the saving quality that while this pressure is on one side of the bearing the oil film is given an opportunity to form on the opposite side and be ready to meet this. On account of the above action that takes place in a large reciprocating-engine bearing we need a fairly heavy oil to give an increased "cushioning" effect and to prevent wasting the oil. The limit of viscosity in forced-lubrication systems is set by the inability of the oil pumps to handle a too heavy oil. With engines not fitted with forced-feed lubricating systems it is set by the ability to feed properly.

Turbines fitted with forced-feed systems give the nearest approach to perfect lubrication. In case the bearings heat up it may be due to grit getting in the bearing, and in this case the pressure on the system can be increased with the hope of forcing this out. The increase in pressure does not lessen the friction, though it does assist in carrying off the heat generated by friction and keeps the bearing slightly cooler.

PHYSICAL CONSTANTS.

As so much attention is often paid to physical constants, it might be well to make a few remarks here concerning them.

Viscosity.—We have already discussed this property and the effect of varying conditions on it; it only remains to describe how it is measured. The standard method is to express it as the number of seconds of time required for a given amount of oil to flow through a standard orifice. In the naval service the standard viscosimeter is the Saybolt, and the viscosity is generally expressed as so many seconds Saybolt at 100 F. Forced-lubrication and turbine oils, for instance, vary in viscosity from the lightest oils, which have a viscosity of about 160 seconds, to the heaviest, which have about 750 seconds at 100 F.

Flash Point.—This is the temperature at which vapors are given off in sufficient quantity to give a momentary flash when

a light is passed across the surface of the oil. For ordinary lubrication it is not of much value as long as it is high enough to be safe. Practically all lubricating oils have a flash point above 300 F. For the lubrication of steam cylinders a high-flash oil is necessary. For motor-cylinder lubrication a 300 F. flash point should be sufficiently high. This point was discussed at length in an article "Motor Cylinder Lubrication" in the February, 1915, number of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

Fire Point.—This is the temperature at which vapors are given off in such quantity that the oil will catch on fire and continue burning if a light is passed across its surface. Roughly it is generally about fifty degrees higher than the flash point. Except for this difference, its value as indicating anything about an oil is the same as that of the flash point.

Specific Gravity.—This gives no indication of the lubricating value of an oil. To the expert it will give some indication of the nature of the crude from which it was obtained. Many consumers, however, still buy their oil on specifications as to gravity with the mistaken idea that this is an index of its lubricating value.

Color.—The color of an oil gives no indication of its lubricating value.

Cold Point.—This is the temperature at which an oil becomes too viscous to flow. An oil should, of course, have a cold point much lower than the temperature of the room in which it works.

KINDS OF OIL FOR DIFFERENT PURPOSES.

The general qualifications of different kinds of oil that are in use in the naval service are as follows:

Marine Engine Oil.—This kind of oil is used on marine engines without forced lubrication, it being fed either by wick or sight feed. As the oil is only used once, it must be heavy enough to stay in the bearings and not be squeezed out easily

—otherwise excessive waste will take place. Also, since water often gets in the bearings or is splashed on the crosshead slides, some provision must be made to prevent the oil from being washed away from the bearing surfaces.

On account of the foregoing reasons, marine-engine oils are composed of mineral oils compounded with a certain percentage of animal or vegetable oil—generally about 15 to 20 per cent. of blown rape-seed oil. This is the only kind of oil used in the Navy that is not a pure mineral oil. Animal and vegetable oils maintain their viscosity with increase of temperature much better than mineral oils; also when stirred with water they form thick emulsions, or, to use the usual engineering term, they "saponify." The effect of this saponification is to thicken the oil and to make it more adhesive. On that account a film of mineral oil can be washed off of a crosshead slide much more easily than a vegetable or animal oil.

Tests are now being carried out at the Engineering Experiment Station to determine if it is not possible to obtain a straight mineral oil that will answer this purpose, as many of these oils emulsify very readily. This would reduce the price of this class of oils considerably.

Forced-Lubrication Oils.—This class includes ***turbine oils,*** the difference in the two being one of viscosity only. Conditions in forced-feed systems are quite different from those in sight or wick-feed systems. Since the oil is used over and over again, any that may be squeezed out from between the bearing surfaces does not represent a loss; also the oil can be supplied in any amount, thereby eliminating the possibility of an insufficient supply to the bearing. Therefore a lighter oil may be used than with the wick feed.

Water may get in the bearings, as mentioned before, but in this case any oil that is washed off will immediately be replaced by a fresh supply. There is no need of any "saponification" of the oil here. On the contrary, every effort should be made to exclude water, since the oil must be kept in con-

tinuous circulation, and the effect of a water leak soon becomes cumulative. One of the primary requisites of a forced-feed oil, then, is that it must not saponify and must separate readily from water. The viscosity of the forced-feed oils in the service range from about 150 seconds to 750 seconds Saybolt at 100 F.

Motor Cylinder Oils.—This class of oils was discussed at length in an article "Motor Cylinder Lubrication" in the February, 1915, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

Steam Cylinder Oils.—In steam cylinders the oils are subjected to the high temperatures of the steam, and if their flash point is not sufficiently high they will vaporize, and therefore cannot adhere to the cylinder walls. Also their viscosity must be great enough to withstand this high temperature without their getting too thin. A cylinder oil should, therefore, be selected with reference to the pressure and temperature that will be encountered.

When wet steam is used there is always some condensation on the cylinder walls, and this is liable to wash the oil off if a pure mineral oil is used. To prevent this it is usual to add a small percentage of tallow or lard oil to the cylinder oil to cause it to emulsify and stick to the surfaces similar to the action of a marine-engine oil. The objection to this is that the fatty oil (tallow) will get in the feed water through the condenser and liberate fatty acids in the boilers, thereby causing increased corrosion and scale. On this account only pure mineral oils are allowed to be used in steam cylinders in the naval service. In fact, practically no cylinder oil at all is used in the steam cylinders in the naval service, the wet steam itself furnishing what lubrication is necessary.

For superheated steam no condensation on the walls is available for lubrication and the need of oil is therefore much greater. There is no danger of the oil washing off, however, for the same reason, and a straight mineral oil should be satisfactory. The temperature of superheated steam is, of course,

higher than that of wet steam and a higher flash point is necessary.

Ice-Machine Oil.—Most of the ice machines in the naval service are of the dense-air type. Oil used in the expander valve chest and cylinder should have a very low cold point and should not have too great a viscosity at the low temperatures at which the expander cylinder works. In order to get these qualities it is necessary to use a light oil.

In the compressor cylinder and valve chest the temperature is higher. In some cases it is so much higher that ordinary ice-machine oil will not have sufficient viscosity. In a case of this sort a heavier oil should be used in the compressor cylinder and valve chest. Where the return air is cool enough, regular ice-machine oil can be used in both expander and compressor. It is a good plan to do this where possible, as there will, of course, be some leakage of the oil from the compressor side to the expander side and a heavy oil would be very likely to gum there.

For other working parts of the ice machine the rubbing speed is low and the pressures not very great. Marine-engine oil or forced-lubrication oil should be used with the best results.

Dynamo Oil.—As the load on the bearings of a dynamo is generally light and the rubbing speed fairly great, a light oil similar to a turbine oil should be used. If too large a quantity is used the amount can be reduced by using a slightly heavier oil—for instance, a forced-lubrication oil.

Air-Compressor Cylinder Oil.—Steam cylinder oils give the best results in air-compressor cylinders. Owing to the high temperature and compression, an oil of high viscosity is necessary, and this condition is best fulfilled by steam-cylinder oils.

WATER IN BEARINGS.

A good deal of misunderstanding exists on the subject of water in bearings. A mixture of oil and water will not give as good lubrication as the oil alone. This is shown graphi-

cally in the curve, Fig. 4, which was obtained by measuring the coefficient of friction of a marine-engine oil first alone and then with various percentages of water added to it, the mixture being stirred thoroughly to make it saponify. This curve shows that even small amounts of water in a bearing will greatly increase the friction.

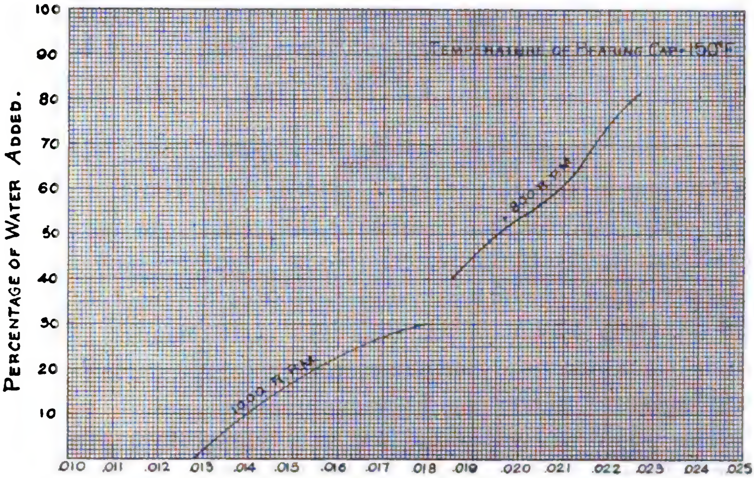


Fig. 4.—Coefficient of Friction.

This seems directly at variance with the old theory that a "lather" on a bearing is a sign of good lubrication. This theory was based on the fact that there was always a probability of water getting in the bearings and a corresponding fear that this water would wash the oil off the bearing surfaces and destroy the lubrication entirely. As froth on the bearing indicated that the oil and water would emulsify, this relieved the engineer's feelings to that extent.

The best plan is to use an oil that will emulsify and then keep water out of the bearings, remembering that if it does get in accidentally no great harm will be done. In cooling off a hot bearing the water should be applied through the water service and on the outside of the bearing, and care should be

taken not to mix it with the oil, as the cooling effect will be more than counteracted by the additional friction.

Forced-Feed Systems.—The foregoing remarks apply principally to sight and wick-feed lubrication. In forced-feed lubrication conditions are different, as previously explained, and emulsification is always objectionable. In addition to increasing the friction and giving less lubrication the water corrodes the journals, and the rust thus formed gradually accumulates in the oil, causing additional wear on the bearings. The so-called "dirt" that accumulates in the oil is principally iron rust.

For this reason it is extremely desirable to get rid of, as soon as possible, any water that does leak into the system. The only practical way is to heat the mixture in the settling tank and draw off the water. Some engineers have an idea that the oil should be allowed to cool after this heating before the water is drawn off, but this idea is erroneous. The settling out is practically all done while the mixture is hot, and waiting for it to cool has little or no effect.

The water will settle out best at a temperature just below the boiling point of water. If 212 F. is exceeded steam will form, and this will serve to agitate the mixture. A temperature of 200 F. should not have any bad effect on the oil. The oil should be cooled before being supplied to the bearings, however.

If marine-engine oil gets in the forced-lubrication system there is a tendency to form a thick, heavy emulsion which gives great difficulty in settling out. Some cases have been known where tallow, which had been put in bearings when a ship was laid up, was allowed to get in the forced-feed oil, and this resulted in the formation of a heavy emulsion which would not settle out.

Every effort should be made to get rid of the dirt in the oil and the strainers should be frequently cleaned. If water is kept out of the system most of the dirt will be automatically eliminated.

GREASES.

For the lubrication of piston rods or steam cylinders pure mineral steam cylinder oil should be used, and even then care should be exercised to keep it from getting in the condensers and boilers. Greases should never be used, as they will liberate fatty acids which eventually get in the boiler.

Greases consist essentially of a mineral oil thickened with soap. The lubrication is given by the oil and the soap simply makes it more solid and gives it greater body. They are used principally where it is difficult to keep an oil in a bearing from being squeezed out or where it is not practicable to give a steady feed to the bearing. They are also used to advantage where the frictional heat of a bearing will melt the grease and supply it automatically. It is a good policy to use straight oil wherever it can be done, however, and only use grease where oil cannot be used.

GRAPHITE.

Graphite is classed as a lubricant because it reduces friction, but its action is different from that of oils. If a highly-polished bearing surface is examined under a high-power microscope it will be seen that the so-called smooth surfaces are really quite rough, they really consisting of alternate ridges and depressions which are too small to be detected with the naked eye. If graphite is used on a bearing of this kind it will soon fill up all the depressions and give a bearing surface that is much smoother. This will, of course, reduce the friction. A steady supply, however, would soon clog the bearing.

Graphite and oil are often used together and will give good results where the graphite does not accumulate and clog the bearing. It often happens that the oil holes or pipes get stopped up in this way. The same objection applies to graphite grease.

U. S. S. *CUSHING.*

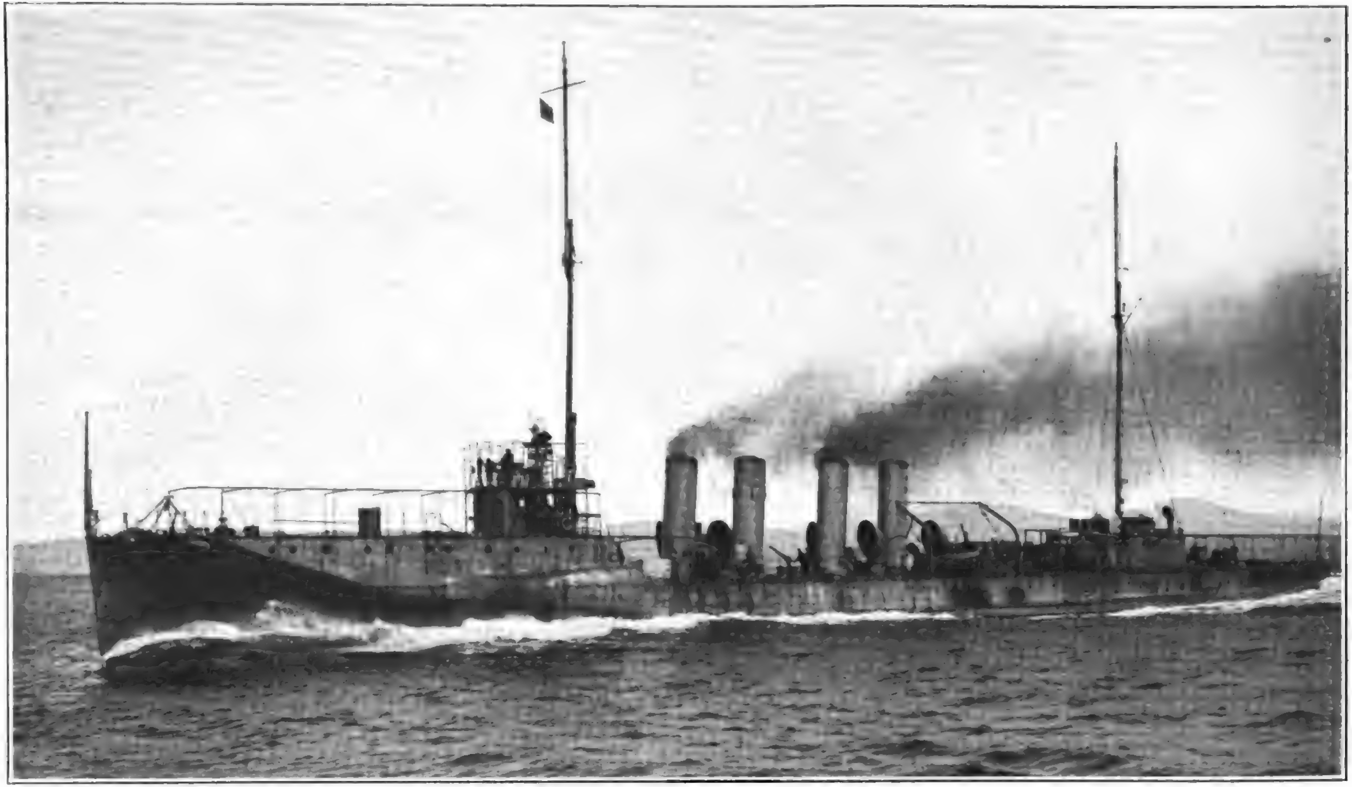
DESCRIPTION AND TRIALS.

BY LIEUTENANT ORMOND L. COX, U. S. NAVY, MEMBER.

Torpedo-Boat Destroyer No. 55, the *Cushing*, is one of the six destroyers of the same class authorized by an Act of Congress approved August 22, 1912. The *Cushing* is a twin-screw vessel fitted with Curtis turbines, with geared cruising turbines and designed for a speed of 29 knots at about 1,050 tons trial displacement with the main engines developing about 16,000 shaft horsepower. The vessel was built under contract by the Fore River Shipbuilding Co. of Quincy, Mass. The contract was signed December 11, 1912, the price being $854,500.00, and the time of construction twenty-four months.

PRINCIPAL HULL DIMENSIONS.

Length between perpendiculars, feet and inches	300–00
on L.W.L., feet and inches	300–04
overall, feet and inches	305–03
Breadth, extreme, over guards, feet and inches	31–01
on L.W.L., feet and inches	30–03
Depth, molded, at side to main deck, feet and inches	17–01
Draught to L.W.L., feet and inches	9–05½
Mean trial displacement, tons	1,050
Ratio, length to beam	9.93
beam to length	.1007
Ton per inch immersion at L.W.L.	14.5
Area midship section, square feet	198
L.W.L. plane, square feet	6,040
wetted surface, square feet	9,420
Coefficient of fineness, block	.4276
midship section	.6924
L.W.L. plane	.6648
cylindrical	.6180



U. S. S. "CUSHING."

Capacity of fuel-oil tanks, tons	346.6
reserve-feed tanks, F.W., tons	18.6
fresh-water tanks, F.W., tons	14.6
cofferdam A-16, F.W., tons	18.3
trimming tank A-1, S.W., tons	16.15
D-8, S.W., tons	9.2

The arrangement of the decks, quarters, machinery spaces, storerooms, oil-storage tanks, etc., is in general the same as for other destroyers of this class, Nos. 51 to 56.

MACHINERY.

The propelling machinery consists of Curtis turbines arranged on two shafts as shown on Plate I. On each shaft there is a 30-inch pitch diameter geared cruising turbine connected to the main shaft by means of a Metten clutch, and a 63-inch pitch diameter ahead and astern turbine in a common casing.

The main turbines are designed to develop about 8,000 S.H.P. each when making about 550 revolutions per minute. The cruising turbines are designed to develop about 450 S.H.P. each at 2,000 revolutions per minute and are geared down in the ratio of 7.95 to 1. The cruising units may be used at all speeds ahead below about 16 knots and are disconnected for higher speeds ahead and while reversing.

The main ahead turbine contains forty-one stages, the first four having three moving rows of blades each, and the other thirty-seven being single-bucket drum stages. The 4th stage wheel is forged integral with the drum, the other wheels being of the usual built-up construction.

The reverse turbine consists of one stage with four moving rows of blades and a drum with nine single-bucket drum stages and is capable of developing about 50 per cent. of the ahead power.

The general construction of the turbine is the same as for previous destroyers built by the same company, with the exceptions that the casing for the 1st and 2d stages ahead is cast steel instead of cast iron and all blades over 6 inches in length are soldered at the root.

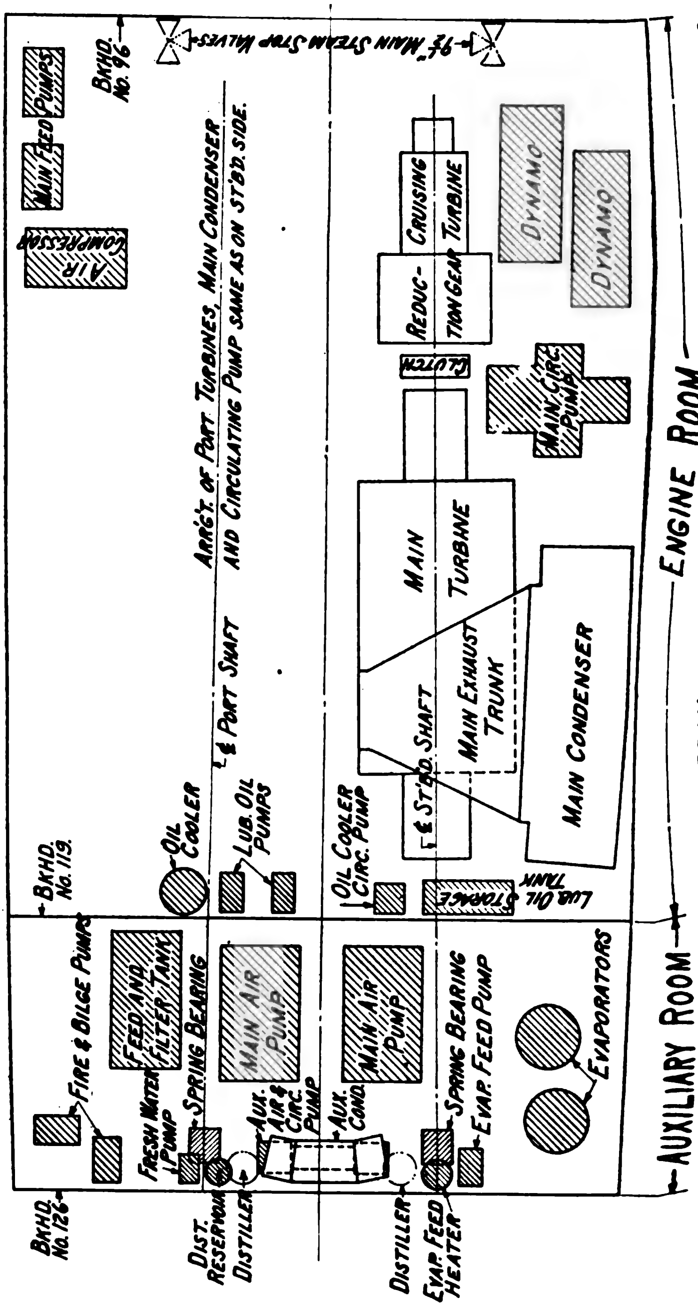


PLATE I - U. S. S. CUSHING - ARRANGEMENT OF ENGINE AND AUXILIARY ROOMS.

All blades, shrouding and channel bars are rolled brass.

On each shaft there is a 30-inch pitch diameter Curtis turbine connected through gearing and a hydraulic clutch to the main shaft. It consists of seven wheel stages, the 1st stage having three rows of moving blades and the other six stages two moving rows each.

The number of nozzles per stage, expansion ratio of nozzles and throat area in square inches is as follows:

Stage.	Number.	E. R.	Area.	Height, ins.
		Cruising Turbines.		
1	2	1.088	.908	.70
2	10	......	2.09	.75
3	14	......	2.926	.75
4	19	......	3.97	.75
5	27	......	5.643	.75
6	38	......	7.942	.75
7	55	......	11.495	.75
		Main Ahead Turbine.		
1	20	1.05	10.00	1.0
2	57	......	22.63	1.125
3	88	......	34.94	1.125
4	116	......	46.05	1.125
		Reverse.		
1	10	1.755	6.75	1.5

SHAFTING AND BEARINGS

Cruising-turbine rotor shaft, length, feet and inches	6–02$\frac{9}{16}$
diameter, maximum, inches	06¼
Main rotor and thrust shaft, length, feet and inches	21–06
diameter, maximum, inches	16½
hole, inches	09
Forward line shaft, length, feet and inches	11–04
diameter, inches	10
hole, inches	07
After line shaft, length, feet and inches	20–08
diameter, inches	10
hole, inches	07
Stern-tube shaft, length, feet and inches	25–08
diameter, inches	10
hole, inches	07

Propeller shaft, length, feet and inches	32-03¾
diameter, inches	10
hole, inches	07
Bearings, main, number, each turbine	2
diameter, inches	13
length, inches	18
type	Babbitted.
Spring, number	2
diameter, inches	10¼
length, inches	14
type	Babbitted.
Stern-tube, forward, diameter, inches	10¼
length, inches	26¼
type	Lignum vitae.
aft, diameter, inches	10¼
length, inches	43¾
type	Lignum vitae.
Strut, forward, diameter, inches	10¼
length, inches	31¼
type	Lignum vitae.
after, diameter, inches	10¼
length, inches	40¼
type	Lignum vitae.

The main thrust bearing is at the forward end of the turbine shaft, and consists of four horseshoes of the regular type, babbitted on both faces and cored for the circulation of cooling water.

The turbine is so designed that the steam thrust balances the propeller thrust at a speed of about 27 knots. At higher speeds the steam thrust is the greater, and at lower speeds, the propeller thrust is the greater. The total effective surface of the horseshoes is 295.2 square inches.

Number of collars	5
Thickness, inches	1 1/16
Space between collars, inches	3
Outside diameter, inches	14¾
Inside diameter, inches	8¾

The shaft horsepower developed by the engines is measured by a Gary-Cummings torsionmeter, one for each line of shafting.

The torsionmeter constants were found by calibration of

the actual shafts with the meters mounted on them. The constant for the starboard shaft is 6.79215 and for the port shaft 6.67132.

PROPELLERS.

There is one three-bladed outboard-turning propeller on each shaft. The propellers are of manganese-bronze, cast solid, and the blades are true screw and machined to pitch and polished. The following are the actual propeller measurements:

	Starboard.	Port.
Diameter, feet and inches	7–05½	7–05$\frac{15}{16}$
Pitch, feet and inches	6–06.152	6–06.011
Ratio, diameter to pitch	.873	.864
Projected area, square feet	26.797	26.78
Helicoidal area, square feet	29.67	29.67
Disc area, square feet	43.69	44.00
Ratio, projected to disc	.6133	.6087
helicoidal to disc	.6791	.6743
projected to helicoidal	.9032	.9026
Lower tip of blade above keel, inches	3¼	3$\frac{1}{32}$
Immersion of upper tip of blade at load draught, ins.	20¾	20$\frac{17}{32}$

MAIN CONDENSERS.

There are two main condensers, pear shaped, of the curved-tube type, the tubes being rolled into the tube sheets. They are built up in the usual manner, the principal dimensions being as follows:

Length between tube sheets, feet and inches	14–00
of tubes as fitted, feet and inches	14–09
Thickness of tube sheets, inches	01½
Tubes, number, each condenser	2,532
diameter, outside, inch	00⅝
thickness, B.W.G.	18
Cooling surface, square feet	5,800
Main exhaust nozzle	10 feet, 04 inches by 22 inches.
Diameter, auxiliary exhaust nozzle, inches	07
dynamo exhaust nozzle (starboard only) inches	08
air-pump suction, inches	11
circulating-water inlet and outlet, inches	19¾
drain connection, inches	03

MAIN AIR PUMPS.

Each main condenser is provided with a Blake vertical, twin-plex, single-acting beam air pump, located in the auxiliary room, with a steam cylinder 12 inches in diameter and wet and dry-air cylinders 28 inches each, with a common stroke of 18 inches. The wet suction is 9 inches in diameter and the dry 7 inches, both taking off the common 11-inch suction from the condenser. From the wet-air pump there is an 8-inch discharge to the feed and filter tank.

MAIN CIRCULATING PUMPS.

There is one centrifugal circulating pump for each main condenser, driven by a two-cylinder reciprocating engine. The engine's moving parts are completely enclosed and supplied with forced lubrication by a small plunger pump driven from the engine shaft.

Data for One Main Circulating Pump and Engine.

Diameter of impeller, inches	33
suction and discharge nozzles, inches	19½
steam cylinders (2), inches	05½
Stroke, inches	06
Revolutions per minute	300

There is one auxiliary condenser, located in the auxiliary room. It is cylindrical in shape and of the curved-tube type, the tubes being expanded in the tube sheets.

There are 280 ⅝-inch tubes having a total cooling surface of 267 square feet. The length of the tubes as fitted is about 6 feet 9 inches between the tube sheets. Directly beneath and connected to the condenser is a Blake horizontal, simplex, combined air and circulating pump 6 by 8 by 8 by 7 inches.

The auxiliary exhaust connection to the condenser is 6 inches in diameter.

Feed and Filter Tank.—This is located in the auxiliary room and has a total capacity of 818 gallons, of which 235 gallons belongs to the filter chambers and 583 gallons to the

feed tank proper. The tank has a 11-inch connection for main air-pump discharge, a 3-inch connection for the auxiliary air-pump discharge, a 8-inch feed-pump suction connection and a 4-inch overflow pipe. There is also a connection from the distiller fresh-water pump and 2-inch vapor pipes leading from each filter chamber.

FORCED–LUBRICATION SYSTEM.

The main turbine and thrust bearings, cruising-turbine bearings and gears are fitted with a complete system of forced lubrication. The system comprises the following:

1 300-gallon storage tank.
1 200-gallon storage tank.
1 158-gallon oil-settling tank.
1 70-gallon drain tank.
1 23-inch diameter Reilly multicoil oil cooler.
2 $4\frac{1}{2}$ by 6 by 6 Blake simplex, vertical oil pumps.

Oil is taken from the drain tank by either of the oil pumps and discharged to the various bearings through the cooler. From the bearings the oil drains by gravity into a 10-inch pipe which runs from the forward bearing of the cruising turbine, aft through the bilge near the center line, to the oil-drain tank.

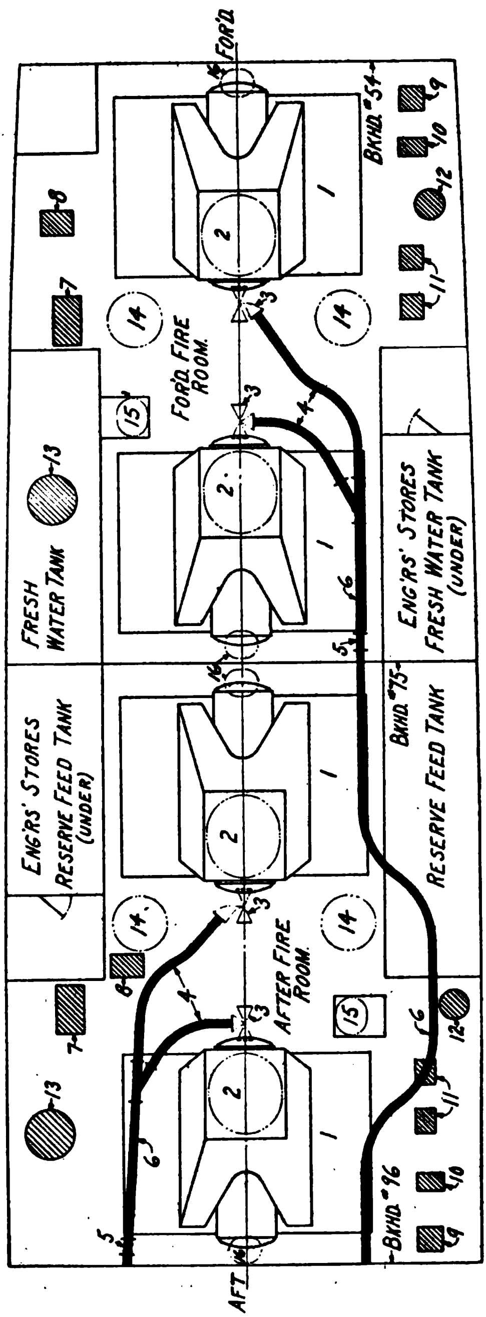
Cooling water for the cooler is supplied by either the main circulating pumps or one of the fire and bilge pumps located in the auxiliary room.

Oil to the clutch is supplied by an independent $4\frac{1}{2}$ by $2\frac{3}{4}$ by 4-inch pump.

The oil cooler has a cooling surface of 73.8 square feet.

BOILERS.

Steam is supplied by four oil-burning Yarrow boilers arranged in pairs in two separate compartments, as shown in Plate II. They are designed for a working pressure of 265 pounds gage, and each boiler has an independent smoke pipe.



ITEM	DESCRIPTION	ITEM	DESCRIPTION
1	BOILER	9	FUEL OIL HAND PRESSURE PUMP
2	SMOKE PIPE	10	" " BOOSTER PUMP
3	7" MAIN STEAM STOP VALVE	11	" " SERVICE "
4	7½" O.D. MAIN STEAM PIPE	12	" " HEATER
5	MAIN STEAM EXPANSION JOINT	13	FEED WATER HEATER
6	10⅛" O.D. MAIN STEAM PIPE	14	TURBINE DRIVEN BLOWER
7	AUX. FEED PUMP	15	AIR LOCK
8	FIRE AND BILGE PUMP	16	ESCAPE HATCH

PLATE II—U.S.S. CUSHING—ARRANGEMENT OF FIRE ROOMS.

The firerooms are operated under the closed fireroom forced-draft system.

Each boiler is equipped with nine oil burners of the Fore River type.

Data for One Boiler.

Pressure, working, pounds per square inch, gage	265
test, pounds per square inch, gage	400
Drum, steam, inside diameter, inches	43
thickness of tube sheet, inches	$1\frac{5}{8}$
wrapper sheet, inch	$0\frac{9}{16}$
water, inside width, inches	$27\frac{11}{16}$
depth, inches	$19\frac{25}{32}$
thickness of tube sheet, inches	$1\frac{1}{2}$
wrapper sheet, inch	$0\frac{15}{32}$
Number of 1-inch tubes	2,656
$1\frac{1}{4}$-inch tubes	218
Heating surface, square feet	5,375
Furnace volume, cubic feet	630
Diameter of main stop valve, inches	7
feed, stop and check valves, one to each lower drum, inches	2
auxiliary feed, stop and check valve, inches	3
Diameter of surface-blow valve, inches	$1\frac{1}{2}$
bottom-blow valves (two), inches	$1\frac{1}{4}$
safety valves (two duplex), inches	4

The auxiliary feed enters the steam drum and the main feed enters the water drums, where it is baffled so that it rises through the two outer rows of tubes.

All tubes are straight except the two rows nearest the furnace, which are curved slightly to provide for expansion. These latter are $1\frac{1}{4}$ inches external diameter, all others being 1 inch.

FUEL-OIL SYSTEM.

This system consists of two light-service booster pumps, one in each fireroom; four duplex service pumps, two in each fireroom; one oil heater in each fireroom, and the oil-storage tanks, valves and piping as required.

The booster pumps draw oil from the storage tanks and discharge to the suction of the service pumps. The latter draw either from the storage tanks direct or from the dis-

charge line from the booster pumps and discharge to the burner lines.

There are nine burners per boiler, each supplied by a ⅜-inch branch from the burner line.

The whole system is completely fitted with strainers, stop, and automatic stop valves.

FORCED–DRAFT BLOWERS.

There are two 35-inch single-inlet Sturtevant cone fans in each fireroom, each driven by a Curtis turbine direct connected to the fan. The turbine has one stage with three moving rows of blades.

FEED–WATER HEATERS.

There is one Reilly vertical, multicoil feed-water heater in each fireroom. The heaters are 36 inches in diameter and each contains 29 coils, with a heating surface of 178.35 square feet.

MAIN STEAM PIPING.

The main steam connection on each boiler is 7 inches in diameter, the forward boiler steam lines combine into one 9½-inch line which runs aft on the starboard side. The two after boilers combine into one 9½-inch line which runs aft on the port side, the two main lines being cross connected with a 6½-inch cross connection. There are expansion joints in each line at the engine-room bulkhead and a stop valve at the engine-room bulkhead. Just inboard of the bulkhead stops there are 9½-inch astern and ahead turbine throttles and a 3-inch throttle for the cruising turbines.

AUXILIARY STEAM PIPING.

There is a 3½-inch stop valve on each boiler for supplying the auxiliary steam line. In each fireroom connections are taken off these lines for the various auxiliaries in the fireroom, the two lines in each fireroom then combine into a common

3½-inch line. In the after fireroom these lines are cross connected and a 4-inch line runs aft on each side of the engine room to the auxiliary room, where they are cross connected, forming a loop. From this line branches lead to the various auxiliaries.

AUXILIARY-EXHAUST PIPING.

The auxiliary-exhaust piping runs throughout the machinery spaces, varying in diameter from 2½ to 8 inches. There are branches to each auxiliary and the following connections:

4½ inches to each feed-water heater.
6½ inches to the 1st and 13th stage of each main turbine.
6 inches to the auxiliary condenser.
7 inches to each main condenser.

MAIN AND AUXILIARY FEED SYSTEMS.

There is an 8-inch suction main running forward from the feed and filter tank with a 5½-inch connection to the air-pump channel ways and a 5½-inch connection to each main feed pump in the engine room. The line runs forward to the after fireroom, where it divides into two 5½-inch branches, one to each auxiliary feed pump in the two firerooms.

The discharge from each feed pump is 4 inches in diameter, the discharges from the two main feed pumps combining into one 6-inch line which runs forward to the firerooms, with a 4-inch branch to each feed-water heater or by-pass, thence to the lower drums of the boilers.

Each auxiliary feed pump discharges direct through the auxiliary feed line to the upper drums of the boilers or cross connects to the main line, thence through the heater or by-pass to the lower drums.

EVAPORATING AND DISTILLING PLANT.

There are two evaporators in the auxiliary room and three distillers in the hatch over the auxiliary room. They have a nominal capacity of 3,750 gallons of potable water per 24

hours, and when clean are to have an overload capacity 40 per cent. in excess of the nominal capacity.

Data for One Evaporator.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	32
Height overall, feet and inches	6-06
Number of coils	10
Diameter of coils, inches	04½
tubes, inch	01
Thickness of tubes, B.W.G	16
Heating surface, square feet	61.5
Diameter of steam nozzle, inches	01¼
vapor nozzle, inches	02½
feed valve, inch	01
blow valve, inches	01½

Data for One Distiller.

Type	Reilly, vertical, multicoil.
Diameter, inside, inches	13
Height between covers, feet and inches	5-05⅝
Number of coils	3
Diameter of coils, inches	04½
tubes, inch	01
Thickness of tubes, B.W.G	16
Cooling surface, square feet	18.5
Diameter of vapor inlet, inches	01¼
drain connection, inch	0¾
circulating-water connections, inches	01¼

TRIALS.

The following trials were required by the contract:

(*a*) A progressive trial over a measured-mile course at Rockland, Me., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four-hours' duration in the open sea in deep water at the highest speed attainable, the average for the four hours not to be less than 29 knots.

Three fuel-oil and water-consumption trials (*c*), (*d*) and (*f*), of four-hours' duration in the open sea in deep water at average uniform speeds of 24, 15½ and 12 knots respectively,

as nearly as possible. The trials to correspond as nearly as possible to service conditions. The cruising units to be used on trials (*d*) and (*f*).

(*e*) An endurance trial of ten-hours' duration in the open sea at an average speed of 15½ knots, made under the same conditions as trial (*d*) in order to test the reliability of the cruising units. No fuel or water measurements were required on this trial.

(*g*) Trials to determine the ability of the vessel to back satisfactorily at full and cruising speeds, time to bring vessel to dead stop from a speed of 29 knots ahead and the distance reached, also trials to test ability to back when going ahead at about 15 knots with the cruising units in use.

In addition to the above trials a trial of two hours' duration under the same conditions as trial (*d*), except that the main turbines only to be used was required.

Fuel Guarantees.—The contractors guaranteed that the consumption of fuel oil per knot run for all purposes, including that necessary for all auxiliaries in use on the trial, should not exceed 700 pounds at 29 knots, 411 pounds at 24 knots, 150 pounds at 15½ knots and 120 pounds at 12 knots.

The standardization trial was run on May 25, 1915, on the measured-mile course at Rockland, Maine. In all 27 runs were made; run No. 22, being unsatisfactory, was thrown out. From the data obtained on the standardization trial it was found to require 569.7 r.p.m. of the propellers for a speed of 29 knots; 437.9 r.p.m. for 24 knots; 266.7 r.p.m. for 15½ knots and 206.7 r.p.m. for 12 knots.

Table II contains the data from which the curves, Plate III, were plotted.

The endurance run was held on July 22, 1915, the other trials required by the contract were held on the dates noted in Table III, which also gives the data obtained.

TABLE I—PUMPS AND CONNECTIONS—U. S. S. CUSHING.

No.	Pumps.	Size (Ins.)	Type.	Suction Ins.	Suction Pipes From—	Discharge Ins.	Discharge Pipes To—	Location.
2	Main Air.	12 x (2) 28 x 18.	Twin, Vertical, Bucket, Single Acting, "Blake."	11 1½ 1½	Condensers. Reserve Feed Tanks. Condenser for Sealing Water.	6	Feed Tank.	In Auxiliary Room.
2	Main Circulating.	33 dia. Runner, 6½ x 5½ x 6 Engine.	Vertical, Double Engine, Centrifugal Pumps.	19½ 7	Sea. Bilge.	19½	Main Condenser.	In Engine Room.
2	Main Feed.	15 x 10 x 16	Vertical, Piston, Double-Acting, Single, "Blake."	3½ 5½	Feed Suction Pipe and Res. Feed Tanks. Cross-Conn. Pipe.	4	Main Feed Disch.	In Engine Room.
2	Aux. Feed.	Do.	Do.	5½ 1½	Feed Suction Pipe and Res. Feed Tanks. Hose Connection.	4 1½	Aux. Feed. Hose Connection.	One in each Fire Room.
2	Fire and Bilge.	7 x 7 x 12	Do.	4 4 1½	Sea. Drainage. Hose Connection.	2½ 3½ 1½ 1½	Fire Main. Overboard. Distiller. Flushing System thru Fire Main. Hose Connection.	In Auxiliary Room.
2	Fire and Bilge.	Do.	Do.	3½ 4 1½ 4	Sea. Drainage. Hose Connection. Independent Bilge.	2½ 3½ 1½	Fire Main. Overboard. Hose Connection.	In #1 Fire Room.
1	Aux. Air and Circ.	6 x 8 x 8 x 7	Combined, "Blake."	4 4	Aux. Condenser. Sea.	3 4	Feed Tanks. Aux. Condenser.	In Auxiliary Room.
1	Evaporator Feed.	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, "Blake."	2 1½	Sea. Distiller Circ. Water Disch.	1½	Evaporators.	Do.
1	Distiller Fresh Water.	3½ x 4 x 4	Do.	1½	Distiller Reservoir Tank.	1½ 1½ 1½ 1½	Res. Feed Tanks. Ship's Tanks. Cofferdam. Main Feed Tank.	Do.
2	Lubricating Oil.	4½ x 6 x 6	Do.	3	Lubricating Oil Tank.	2½ 1	Bearings. Oil Settling Tank.	In Engine Room.
4	Fuel-Oil Service.	6 x 3½ x 8	Vertical, Piston, Double-Acting, Duplex, "Blake."	2½ 2½	Oil Storage Tanks. Booster-Pump Discharge.	2	Burners.	Two in each Fire Room.
2	Fuel-Oil Booster.	4½ x 6 x 6	Vertical, Piston, Double-Acting, Single, Light Service, "Blake."	3 2½	Storage Tanks. Deck-Hose Conn.	2½ 2½ 2½	Service-Pump Suction. Storage Tanks. Deck Hose Conn.	One in each Fire Room.
2	Turbine Drain.	3½ x 4 x 4	Horizontal, Piston, Double-Acting, Single, "Blake."	1½	Turbines.	1½	Filter Tank.	One under each Main Turbine.
1	Oil to Clutch.	4½ x 2½ x 4	Vertical, Piston, Double-Acting, Duplex, "Blake."	1	Tank.	½	Clutch.	In Engine Room.

TABLE II - U. S. S. CUSHING
STANDARDIZATION TRIAL DATA - MAY 25, 1915.

No. of Run	Time on Course		Speed in Knots	R.P.M.			S.H.P.		
	Mins.	Secs.		Star. Engine	Port Engine	Mean	Star. Engine	Port Engine	Total
1	7	12.7	8.32	141.32	148.00	144.66	96	148	244
2	7	30.9	7.98	137.16	143.88	140.52	121	166	287
3	6	50.7	8.77	148.48	154.97	151.73	151	176	327
Mean of Group			8.26			144.36			286
4	5	0.5	11.98	205.09	211.67	208.38	348	410	758
5	4	54.3	12.23	204.80	209.32	207.06	334	363	697
6	5	9.2	11.64	203.43	208.68	206.06	359	390	749
Mean of Group			12.02			207.14			725
7	3	53.2	15.44	257.91	262.89	260.40	613	737	1350
8	4	2.3	14.86	260.86	264.35	262.61	691	776	1467
9	3	48.1	15.78	261.50	265.64	263.57	657	762	1419
Mean of Group			15.24			262.30			1426
10	3	4.0	19.57	352.40	348.55	350.48	1747	1674	3421
11	2	54.2	20.67	353.34	349.36	351.35	1752	1701	3453
12	3	3.2	19.65	354.92	352.19	353.55	1784	1715	3499
Mean of Group			20.14			351.68			3457
13	2	25.3	24.78	437.88	440.49	439.19	3480	3673	7153
14	2	36.1	23.06	434.72	437.45	436.09	3455	3531	6986
15	2	24.3	24.95	437.03	436.57	436.80	3503	3466	6969
Mean of Group			23.96			437.04			7024
16	2	24.5	24.91	480.21	480.11	480.16	4831	4835	9666
17	2	13.8	26.91	480.56	483.27	481.92	4863	4965	9828
18	2	23.2	25.14	481.64	486.06	483.85	4874	4994	9868
Mean of Group			25.97			481.96			9798
19	1	59.4	30.15	571.94	577.06	574.50	7653	7892	15545
20	2	8.4	28.04	570.97	574.31	572.64	7640	7778	15418
21	2	1.0	29.75	558.16	559.74	558.95	7279	7356	14635
Mean of Group			29.00			569.68			15254
23	1	58.8	30.30	586.65	581.81	584.23	8049	7802	15851
24	2	4.1	29.01	599.64	593.89	596.77	8309	8201	16510
25	1	57.7	30.59	598.07	597.46	597.77	8370	8251	16621
26	2	4.5	28.92	602.36	595.96	599.16	8469	8310	16779
27	1	59.7	30.08	578.18	579.96	579.07	7776	7622	15398
Mean of Group			29.68			593.84			16384

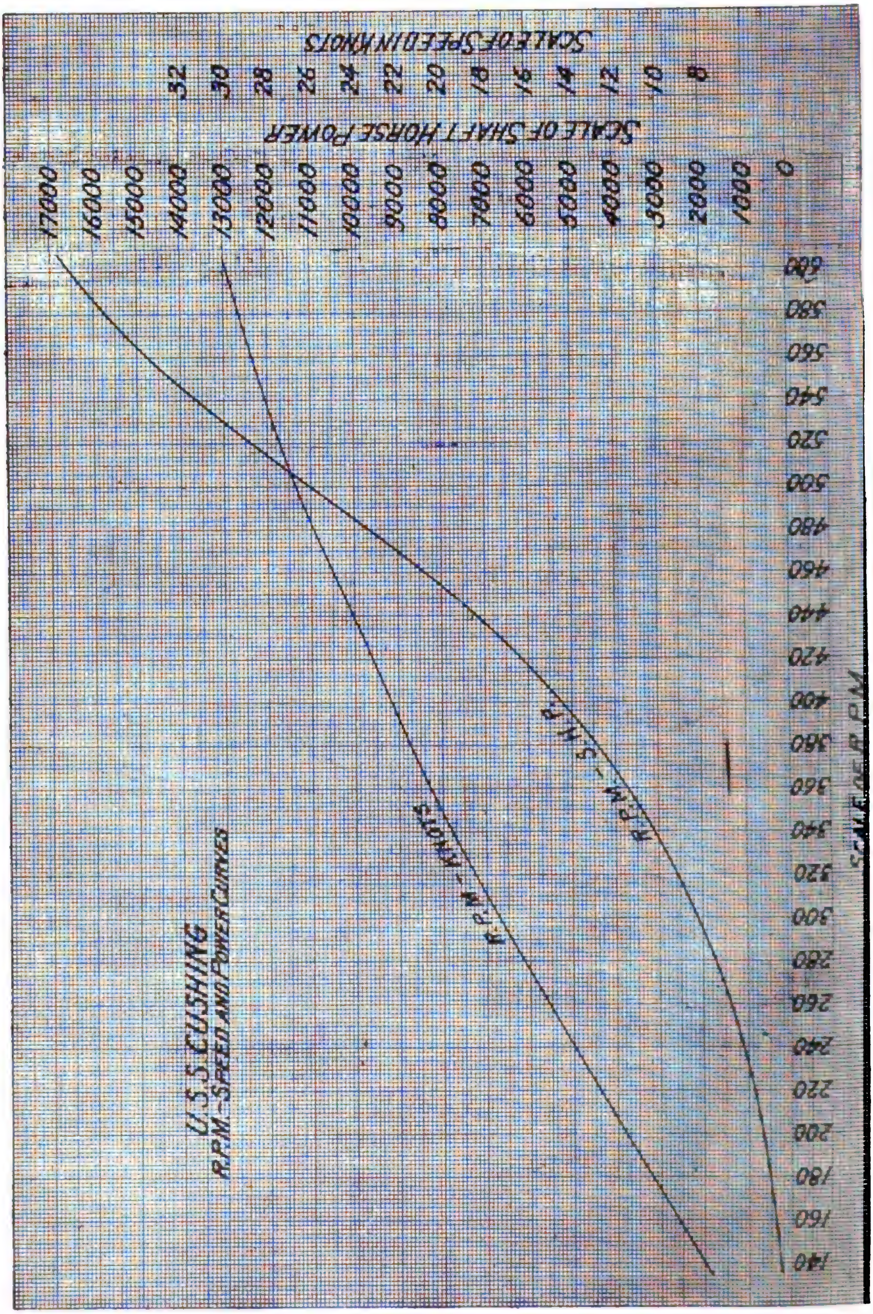


PLATE III.—R.P.M.-SPEED AND POWER CURVES.

TABLE III. TRIAL DATA U. S. S. CUSHING.

	4-Hr. Full Speed Trial	4-Hr. 24-Knot Trial	4-Hr. 15½-Knot Trial	2-Hr. 15½-Knot Trial	4-Hr. 12-Knot Trial
Date of Trial	May 26,1915	May 25,1915	June 7,1915	June 7,1915	June 7,1915
Speed in Knots	29.183	24.178	15.484	15.556	11.988
Draught, Mean, Feet and Inches	9 - 5	9 - 6	9 - 6	9 - 6	9 - 6
Displacement (Corresponding), Tons	1048.0	1054.9	1047.6	1050.4	1046.8
Number of Boilers Used	4	4	2	2	1
from Curve	15575	7290	1495	1520	720
WATER CONSUMPTION, All Machinery					
Lbs. per Hour, Measured	233950.4	125156.2	25869.8	43648.1	13575.5
Reserve Feed	9171.0	2918.0	4790.0	None	5831.0
Evaporated	243121.4	128074.2	30659.8	43648.1	19406.5
" per Sq.Ft. H.S.	11.31	5.96	2.85	4.06	3.61
Lb. of Oil	12.46	13.66	13.20	13.20	13.57
per S.H.P.	15.61	17.57	20.51	28.72	26.95
Knot Run	8331	5298	1980	2806	1619
OIL CONSUMPTION					
Lbs. per Hour	19613.6	9379.2	2323.4	3306.4	1429.8
per S.H.P.	1.259	1.287	1.554	2.175	1.986
Sq.Ft. of H.S.	.9120	.4362	.2161	.3076	.2660
Knot Run	672.1	387.9	150.0	212.5	119.3
Specific Gravity at 60°Fahr.	.9213	.9213	.9135	.9135	.9135
B.T.U. per Pound	19298	19298	19226	19226	19226

*Note.- S.H.P. used is that from Standardization Curve.

THE RESERVE FORCES OF NAVAL MATERIAL.

COÖPERATION BETWEEN THE NAVY AND THE PRODUCERS OF NAVAL MATERIAL.

BY H. C. DINGER, LIEUTENANT-COMMANDER, U. S. N., MEMBER.

The reserve forces of our naval material are represented in the manufacturing plants that produce the material from which the ships, guns and machinery are built and also the commercial shipping which may be converted into naval auxiliaries.

A country without these reserve forces well developed, well founded, prosperous and progressive is at an incalculable disadvantage in comparison with one that, if need be, can put a second fleet into being.

The personnel connected with the production of naval material is also a connecting link between the naval service and the great body of the people. In developing our capacity to manufacture and produce naval material, our naval power is strengthened and developed and is placed on a real sound foundation. A country incapable of building its own vessels and manufacturing its own naval and military equipment is deficient in naval resources and is bound to be at a disadvantage to one that is capable of sustaining and reproducing its own fleet.

It is, therefore, essential that in considering the development of our naval forces the efficiency and well being of the naval reserve of material receive not only attention, but a fostering care.

The ability to build ships and machinery, manufacture guns and armor, is an asset to our naval strength, no matter whether this is in civilian hands or under direct control of

the Navy Department. The commercial dry dock capable of taking the largest naval vessel is as much real assistance to the Navy in time of war as is one at a navy yard. A commercial repair plant that can rehabilitate a damaged dreadnaught, or an ordnance shop that can replace damaged armament, is no small factor in enhancing naval power.

From the standpoint of efficiency and economy, the manufacturing done by the Government should be an adjunct to the manufacturing facilities of the commercial plants of the country. The commercial plant designed to produce naval material can, to a large extent, be utilized also for other than naval work, hence such a plant may bring some return for its cost should naval work fall behind. To increase the facilities for manufacturing naval material much beyond the demands of the Navy building program is a serious economical mistake. It makes no great or material difference to the country at large whether the waste of its manufacturing plants is made by the Government or by the private firm. If there is over production of these facilities, a portion of the plants will lie in idleness and the country loses that much. If the plants are in private hands, they may be utilized to some extent for commercial purposes, but there is still the economic loss.

The Government plant has no reason to be opposed to the private plant, and the best final result both to the taxpayers and to the personnel interested in the manufacturing will be produced when Government establishments and the various commercial plants producing military material operate as properly adjusted component parts of the naval engineering facilities of the country. An over supply of these facilities, which might engender ruinous competition and serious antagonism among the different firms, is of no real benefit to the country, any more than is a ruinous railroad rate war. In a ruinous competition it is very often the labor of the organizations interested that suffers. The loss must be made up somehow and someone must pay for it. The best conditions will, therefore, be produced when there is an adequate amount of naval work to keep all of the plants equipped for this

purpose provided with sufficient work to keep them reasonably employed.

To secure progress, improvement, proper and healthy competition, there should be several firms in the field for any class of work. A monopoly is bad for both progress and cost.

The ideal division of work is such as will keep all of the plants at hand reasonably supplied with work. The total facilities provided, whether under Government or private control, should be proportioned to handle the work. If a definite military and naval building policy were present, the facilities might be properly adjusted so that the plants of the country would have enough but not too much capacity and that they might plan ahead so that they would be most efficient in their equipment and organization. Under these conditions the product could be produced for the least actual cost to the people of the country, and under such conditions our manufacturers would be in the best possible position to cope with foreign competition, and work for foreign governments might then be done with adequate return and as a desirable business venture. To have one plant take work away from another by reason of a slight difference in cost is of no benefit to the people at large, and to have half of the plants idle is decidedly bad for the country. To have all the plants reasonably occupied at a reasonable price for the product is the most desirable result.

The Government can, of course, do a great deal to produce the most satisfactory conditions, but it can not do everything. The governing influences of the various private firms must be willing to coöperate not only with the Government but with each other. The shipbuilder must coöperate with the gunmaker, the steel mill and the founder. It is desirable to have each of the subdivisions of the activities that produce naval material highly specialized in order that the most efficient plant and labor for each part may be present.

The producers of naval material and the Government have a common ground on which to stand and can have a common end in view. There should be healthy competition between

rival firms, but this should not be antagonism. The ability, brain and initiative of the engineering talent should be exercised to develop a better and fitter product, this being the surest means for beating a competitor.

The activities of the Government representatives should be to stimulate these conditions of coöperation, mutual understanding and the furnishing of all possible incentive to the production of the most improved, fit and reliable product that the brains of the American engineer can design and the handicraft of our artisans fabricate. The Navy of this country has developed a field for the inventive genius of the country's engineering talent. It is the largest single customer of the engineering trade. The standards of engineering performance are a guide often used by others. Full, free and open coöperation and a recognition of the mutual interests that exist between the producer and his customer make for better satisfaction to both.

The Navy relies on the commercial engineering field for the excellence of the products from which the material matters of our naval force are constructed. It relies on it for the development of tools, methods of work and the training of artizans by which our fighting weapons are produced in superior form and efficiency. Without a high state of engineering ability and progrees in the country at large, the highest character of excellence in navy material can not be realized. The capacity of our commercial engineering plants is the principal asset of our naval engineering reserve of material.

The promotion among the civilian forces engaged in the production of naval material of a feeling of coöperation, a community of interest and a general feeling of fealty to the end in view—that is, the production of the best possible Navy on the appropriation provided by Congress—is a field for development in which the naval service and all those connected with the supply and manufacture of naval material should be particularly concerned.

Harmonious relations between the naval official and those

in the commercial world supplying naval material are essential to the production of the best naval material, and should be one of the essential policies of the Government. The military forces of the Government are for the service of the public, and should be produced in the greatest possible volume for the effort expended. The commercial plants producing naval material are an adjunct to our military preparedness, and should receive such care and treatment by the Government as will put them in the best condition to assist the Navy.

The business relations existing between the Navy Department and the producers of naval material and supplies are an important matter affecting naval administration, and too often this fact is lightly regarded by those of the Navy, by our legislators, by the manufacturers concerned and by the public at large.

A better understanding beween the customer and purchaser whereby a better knowledge of requirements and a better coöperation is arrived at is bound to bring about better conditions and advantage to both.

It is extremely desirable that any prejudice on the part of manufacturers against the Navy work be overcome. Some believe that such prejudice exists, and by reason of it and by reason of some of the conditions said to exist, the Government is not obtaining the most efficient service that might be secured from the sums expended.

Improvement in this field will, therefore, operate as an important means for securing economy and higher efficiency in our naval expenditures.

The two most important matters to aid in improvement of naval business methods are:

(1) Remove all possible causes for unnecessary prejudice, annoyances, delays, misunderstanding, etc., on the part of manufacturers.

(2) Give the best possible publicity and accurate information concerning naval business methods in order that prejudice due to lack of information and understanding may be eliminated as far as possible.

GOVERNMENT PURCHASING BUSINESS.—NECESSARY FORMALITIES AND PROCEDURE.

All Government bodies are governed by laws and regulations, many of them quite undesirable and vexations to the Government officials who have to comply with them and which seem to the Government contractor in many cases a useless mass of red tape. These laws, it may be admitted, are in some cases and, to some extent, ill-advised and make Government business, to some extent, more dilatory and cumbersome than would be necessary in private business.

As long as the laws exist they have to be complied with, and the manufacturers and business men interested in Government work have a far greater opportunity to remedy these laws than have the Government officials who are often blamed and ridiculed for delighting in red tape which they would gladly cut adrift from should the law permit.

Most of the laws, however, have (or have had) a reason for their being, though sometimes the reason may not be deemed meritorious from a business point of view, as, for instance, the eight-hour law, the law that contracts are not to be let to firms engaged in restraint of trade, etc.

It must be recognized at the outset that a Government Department can not do business like a private business man. If this fact is recognized, it may also be accepted as a fact that it is practically impossible for a Government (owing to the nature of its organization and the various restrictions placed upon its discretion in action) to do and conduct business in as efficient and expeditious a manner as can a private business properly organized and administered. A Government Department, other things being equal, is initially handicapped, and on account of the conditions prevaling can not be as economically managed as could a private firm, with the identical personnel. However, by efficient system and wise administration better conditions may be secured and, above all, better coöperation with the producers can be effected. Also by proper effort, bad laws and restrictions can from time to time be removed or modified.

The aim of the Government should be to award contracts in a manner that will secure the most efficient and suitable material at a profit to the manufacturer that will enable him to give (1) fair and just working conditions to his employees, (2) a stimulus to improve his product and (3) to deliver the best possible work, not merely work that can just be slipped through. To do this, unrestricted, free and antagonistic competition is not the remedy, but, on the contrary, is subversive to it. Free competition carried to its logical end naturally results in a cheapening and a deterioration of the product and, from a business and economic point of view, is illogical and inefficient. It is unfair to the people at large and benefits no one except, possibly, some speculating contractors. To require that Government contracts should go without reserve to the lowest bidder is very bad. It is an easy way to let the contract, but it is about the most inefficient one that readily suggests itself (barring favoritism and dishonesty).

For free and unregulated competition there should be substituted a restricted and discretionary competition which would allow the Government to properly make use of other desirable considerations besides the lowest price. This should be arranged so that the responsible Government official might more nearly conduct the making of purchases along the lines used by the reliable and efficient private firm.

This can to a great extent be done in accordance with existing laws, wherever the responsible Government official is willing to take the responsibility in an endeavor to get the best results and not merely follow the beaten pathway of easy precedent.

Restricted competition would consider (1) that the contractor is a bonafide manufacturer or his direct representative; (2) that the contractor has the proper financial backing, reputation, plant and experience that will insure his being able to fulfill his contract; (3) that the official authority is empowered with a certain amount of discretion which takes into account the relative quality, suitability and adaptability of

the product; (4) for special material, limited to naval uses, work should be divided to some extent among at least two firms in order that the facilities of these plants may be kept in efficient condition and enable competition as to price and improvement of quality to be developed. An equitable manner of doing this is to give a larger share to the lowest bidder and a smaller share to others at the same price, or to limit the amount that may be supplied by one firm. The above should be done for such things as aviation equipment, armor, armament, ammunition and special instruments of type designed especially for naval use.

There should be a building up of ability and facilities to compete for business as well as the use of competition to obtain reasonable and fair prices.

New Contractors.—A new contractor is very likely to fall into the fatal error of trying to deal with the Government in the same manner as he has been accustomed to do with an old customer down the street.

Instructions may be sent and clearly stated as to how business is to be done, but these are often disregarded.

A common fault is for contractors to disregard the local inspector to whom they have been advised to address all communications and write direct to the Navy Department. These communications, after sifting through various offices in the Department, are referred back to the inspector who, in most cases, could have furnished the information immediately by telephone.

Large bodies move slowly, and it is an impossibility for an organization as large as the Navy Department to act on matters with as much dispatch as a firm doing, perhaps, a very specialized business. Especially is this so since the Navy Department is, and has for some time been, woefully under-manned to properly handle, according to commercial business ways, the large volume of work required. A private concern doing about forty millions of purchasing a year would have a specially trained and expert force to carefully handle all the details, but the Navy Department is forced to get

along as best it can on the very limited force which Congress grudgingly supplies. It is saving at the spigot and wasting at the bung-hole, but these are the conditions that our legislators have imposed and we have to make the most of them. To materially improve conditions a more specially trained force is necessary.

Contractors' Unfamiliarity.—A great deal of the trouble is due to dealers and manufacturers being unfamiliar with what is wanted and the methods of doing business. The Navy is one of the largest business concerns in the country, and largely by law is compelled to do business in a certain way, which way is designed to be the best that the laws, conditions and the efficiency and capacity of its personnel will allow. Steps are constantly being taken to improve the system, but in many cases it is cumbersome and apparently complicated. This system can not, however, be changed to suit each new bidder that comes along. Among small firms direct contact and communication between the responsible head of the firm is readily made and verbal decisions can be reached. In such a large organization as the Navy, direct contact with the responsible head is almost impossible, and matters must follow a system of routine, or otherwise there would be utter confusion.

The coöperation between the Navy and the producers of naval material is affected by the manner in which the Navy Department does business with the commercial world, and it is important that it be conducted so that the best possible relations exist. The Navy should handle its requirements to secure its products at as reasonable a cost as possible.

Some objections to the manner with which the Government conducts its dealings are made from time to time. Some of these objections are no doubt well founded, and can be remedied. Others, though objectionable from the point of view of the contractor, can not be removed for various good reasons. A setting forth of some of the reasons that make Government business to some extent undesirable, with suggestions for possible improvement or explanation of reasons for the requirements, may serve to clarify the whole matter.

METHODS OF IMPROVING KNOWLEDGE ON THE PART OF MANUFACTURERS OF NAVAL REQUIREMENTS.

The principal method of improving conditions in this respect is to disseminate information to the producers and provide for having authoritative representatives of the Navy confer with and study the possibilities of the different firms who may be in a position to do work for the Navy.

Considering the amount of work involved, little is done by the Navy Department in placing before the commercial engineering world what its requirements are. It has been the practice generally to wait for the commercial engineer to develop something that he thinks the Navy might want and then allow him to press his argument for its adoption. Sometimes they succeed in having laws passed to put their apparatus in use against the advice of responsible naval officials. This usually is a waste of money. This system works after a fashion, but it would be much better for all concerned if the Navy would make special endeavors to state what it was looking for, what it was desirous of having developed, and what it really needs. If this were done in an efficient manner, much delay and working at cross purposes would be avoided. Desired improvements would be more readily met and unnecessary work on the part of both the Government and the manufacturers would be avoided.

The Navy has in the personnel of its inspection force, scattered over the country, a medium for getting in direct touch with the commercial producers of naval material, and supplying this aggregation of producers with first-hand, accurate and reliable information as to what naval products are wanted and desired, how naval work is handled, and other general information and data concerning it.

The inspection force, if made adequate and efficient, can act as a bureau of information to the possible contractors and as a grand system for collecting information for the Navy Department. It can serve to interest a larger number of manufacturers in any particular field and teach them how they can coöperate with the Navy Department and with other

manufacturers by whom their product may be used to produce some implement for naval use.

At present there are inspection forces under the various bureaus of the Navy Department as follows:

Inspectors of Machinery and *Inspectors of Engineering Material*, under the Bureau of Steam Engineering.

Superintending Constructors and *Inspectors of Hull Material*, under the Bureau of Construction and Repair.

Inspectors of Ordnance, under the Bureau of Ordnance.

Inspectors, under the Bureau of Yards and Docks, though most of the inspection at place of manufacture for work under this Bureau is done by the Inspectors of Engineering Material.

If steps were taken to better coördinate these services and secure a general uniformity of doing business under all of the different bureaus, much good would result. Steps for doing a great deal in this line are underway and improvement is slowly but surely being made, notably the following: General specifications for material supplied under Bureau of Ordnance, Bureau of Construction and Repair and Bureau of Steam Engineering are issued, and all specifications for material and supplies under these bureaus are passed upon by a Board consisting of representatives of all the bureaus concerned. Separate specifications under different bureaus are avoided and reduced. Common and standardized practice as to inspection is being developed.

The inspection service is, however, far from all that it should be. To begin with, there is not a sufficient number of officers and expert assistants assigned to the work, and hence the work of collecting and distributing information can only be done superficially.

More officers and the services of mechanical experts in various fields could well be utilized to advantage, especially in connection with conferences of various engineering societies where adoption of standards and development of uniformity in engineering practice is considered. They can also coöperate with the work of other Government Departments, such as the Bureau of Mines and Bureau of Standards. This

matter is most important, not merely to the Navy, but to the general improvement of the commercial efficiency of the country. A few thousand dollars spent in the activities of this service will save millions by securing better results from our naval material and would be worth even much more in the consequent improvement of the commercial engineering efficiency of the country. Therefore every possible step should be taken to improve the *inspection* and *engineering information* service of the Navy so as to make it the most efficient medium for coördination and coöperation between the Navy and the producers of naval material.

Attention may be especially called to the fact that the improvement of the inspection service is to the great advantage of the manufacturing concerns as well as to the Government. Such an inspection service can also do inspection work for any other of the Government departments at very little extra expense. (This was done in the case of the two large colliers building for the Isthmian Canal Commission.) The expense of this inspection to the Canal Commission was trivial because a system and force suited to the work was at hand.

The Navy now buys more than any other Government department and has a more thoroughly organized and widely distributed inspection service for inspecting engineering supplies and material than any other Government department. This force, if made a little more adequate and a bit more efficient, could, if desired, do most of the engineering factory inspection work for nearly all of the Federal Government departments.

A further use of this inspection service is in connection with the work of the Bureau of Standards, Steamboat Inspection Service, etc., for which services the Navy inspection officers could gather information, make engineering investigations, tests in the field, etc.

The expense incurred for such work for other departments could readily be adjusted by transfer of credits or appropriations, a matter that is now readily done.

SPECIFICATIONS.

Practically all naval contracts are let on specifications; therefore the specification is a very important matter and one concerning which considerable objection and criticism has been raised at times. Many of these criticisms, it must be admitted, are well founded. Specifications are troublesome things, but the better and more clear they are, the more efficiently do they serve their purpose.

Objections to specifications consist chiefly (1) In having indefinite terms that do not clearly specify. (2) In having statements that an article must be satisfactory to some official, instead of stating a definite requirement. (3) In giving insufficient information. (4) In requiring unnecessarily costly or special requirements for services where the ordinary commercial material will fully answer. (5) In requiring unnecessarily elaborate tests. (6) In requiring special and arbitrary sizes just different sufficiently from commercial sizes to make the matter a special job, at correspondingly increased cost as well as annoyance.

The greatest possible care must be taken in drawing up the specifications, and this is a matter for which the Government is responsible. Better specifications secure better material at lower cost, hence it is to the interest of the Government to have as expert a force as possible to draw up its specifications. In this matter there is room for a great deal of improvement, though continual advance is being made. The whole matter is considerably limited by the capacity and expertness of the force provided. The Government could well afford to provide a larger, more expert, and a better organized system of preparing its specifications. The purchases aggregate about forty millions a year. Better specifications could easily save four hundred thousand dollars of this sum at an expense of ten or twenty thousand dollars.

The difficulties and annoyance caused by defective specifications is immeasurably greater than the slight extra investigation or labor required to draw up a proper one. This fact

can not be too much impressed upon those who draw up specifications.

The aim of the specifications should be to secure the most economical material for the purpose intended and not necessarily the highest grade of material that can be made. Specifications can easily be made unnecessarily exacting and may require a costly material where one of much less cost would answer better.

Specifications should avoid special or unusual requirements wherever possible and should aim to call for the best standard commercial practice. Every special requirement is eventually paid for as an extra by the Government. Specifications should not hesitate to call for exacting requirements where such are essential and where the extra cost is well applied, but they should not call for special requirements, special quality, odd sizes and dimensions where these are unnecessary.

Efficient specifications can only be prepared when there is provided an expert force for preparing same. The forces provided often are not as expert as they might be if higher qualifications and better pay were provided for certain expert Government employees and it were better realized that a little extra expenditure for improving the system of writing specifications and making purchases would be saved many times in producing better material at a lower average cost. A great improvement is possible by making periodic bulk contracts for as large a number of articles as possible, and for small amounts in emergencies permit of purchase in the open market without advertising. For the large contract sufficient attention can be paid in preparing the specifications so as to be able to get the most serviceable article at the lowest cost.

The Navy Department now issues a considerable number of leaflet specifications covering apparatus and material. An index to these specifications is issued quarterly by the Bureau of Supplies and Accounts. Also General Specifications for Material, General Hull Specifications and General Specifications for Machinery are issued. All apparatus coming within the scope of these leaflet or general specifications should be

required, for under these and the number and designation of the leaflet or general specifications must be clearly stated. If for any reason some departure from the leaflet specifications is desirable, the leaflet should be referred to and the depature or modification clearly stated.

Though all this is required by instructions in force and indicated by common sense, many requisitions and schedules are prepared which are very defective in that they are too general, leave important matters in doubt and do not give sufficient information to the prospective bidder of what is required. The fact that specifications are loosely drawn causes the manufacturer to make an inexact and indefinite bid, or he may bid on something that he really does not actually intend to supply, thinking that, on account of the loosely worded or defective specifications, he may be able to have his product accepted.

DIFFERENCES AND DIFFICULTIES.—WAIVERS.

It has been suggested that matters of this kind should be settled by a Board of Adjudication. This, in most cases, is entirely impracticable and would involve endless red tape. The details of all requirements should be clearly defined in the specifications and contract. Every effort to have inspections, tests, etc., follow promptly should be made. In special instances an appeal to the Bureau concerned can be made, and for good and sufficient reasons certain requirements may be waived.

In the case of contracts for vessels and for other very large contracts, Boards on Changes are provided.

The different bureaus under which material is purchased aim to be reasonable, and in all cases where a contractor has a bonafide and just claim for consideration a proper hearing is given. The Government can not, however, grant every request for waiver that is made nor can it accept responsibility for contractors' mistakes or their lack of knowledge or lack of ability to carry out the contract. There are also some

contractors who will take advantage of the Government should occasion offer.

Jobbers.—Most difficulties are experienced with jobbers who bid without full knowledge, and at the time of the bid have no definite knowledge where they will secure the apparatus or material for which they bid.

Another source of trouble is the manufacturer who does business on his reputation and who considers such practices as requiring a bond or inspection of his product as an insult to his business reputation.

A frequent excuse in case of rejection for non-compliance with contract is that the material is the same as was previously accepted. This is, of course, no reason at all. If the contractor knew his product did not comply strictly with specifications, this should have been stated in his bid, and proper consideration could have been given to the deviation; but to go ahead with the intention of supplying something that does not conform strictly to the specifications because it has previously been accepted is treading on dangerous ground, and though it may go through in some cases, it is very likely to be held up.

IGNORING LOCAL INSPECTOR.

A matter that sometimes causes more or less delay and extra correspondence is the practice, previously touched upon, of some contractors to ignore the local inspector and take up a question direct with the Bureaus at Washington. Such a course merely delays action, since extremely seldom is any action taken without referring the matter back to the inspector concerned.

PAYMENTS.

The Government system of making payments is not difficult to understand if the rules and regulations concerning same are studied. Payment is made on receipt of evidence of service rendered. The supply and forwarding of this evidence sometimes takes considerable time, and this is where most of the delays are caused. Sometimes, too, the

inspectors or authorities receiving the supplies or material fail to report inspection and receipt promptly. In order to have all proper vouchers which are required by law and by the auditor's rulings, somewhat more paper work is required that in most large commercial concerns. The system of payment at present is about as simple as the conditions will allow, and where delays in payment are made it is due to the fault of either contractors or some of the Government officials, through whom the papers are required to go.

The conditions of the payment are stated in the contract, and contractors in making their bid can suggest such conditions as may appear desirable to them, and *such suggestions are considered.*

Partial payments can usually be provided for in contracts involving large amounts, or where there is a long time in delivery.

If the conditions as to payment are definitely stated in the contract, no difficulty or delay in making of payments is presented, but in some cases this point is not clearly brought out in the contract and sometimes as unnecessary hardship is brought upon a contractor.

This matter will in all cases be clear and plain if the *authority* drawing up the requisition or specification will state how payment is to be made and if the *contractor* in making his bid will call attention to any lack of clearness on this point and state terms of payment desired when his bid is submitted.

Difficulties or hardships as to payments can only result due to ignorance or neglect, either on the part of contractor or some Government official, and is not due to the approved method or system of making naval payments.

INSPECTION.

The system of inspection and severity of it have a potent effect.

The final aim of all inspection is, of course, to insure that the contract requirements are met. The inspection should

accomplish this with the least possible cost or delay, both to the Government and to the contractors.

Inspection is directly unproductive, and for that reason must naturally be limited as much as possible consistent with insuring and ascertaining that all requirements have properly been met.

The system of providing inspectors and their assistants is all important. It is a matter that has not received the full consideration that its importance warrants. It is a special class of work, and officers assigned should be well fitted for it, both by experience and professional training.

Inspection is largely a case of a stitch in time saves nine. Of course, if we want to go ahead on the nine-stitch basis no inspection is necessary.

The following are some inspection difficulties known to have taken place:

Material once Inspected at Point of Manufacture Held up at Point of Delivery.

This happens occasionally and has given contractors, at times, reasonable cause for complaint.

These matters are annoying things to contractors as well as to the Government bureaus concerned and evince a lamentable ignorance on the part of the authorities at the point of delivery of requirements of contracts, the specifications and fair dealing. It is the announced policy of the Department that articles inspected at point of manufacture can not be rejected on matters passed by the inspector at the point of delivery, but may be rejected for other defects. It is also the announced policy that the Government can not legally call for more than the contract requires. This seems to be a very difficult thing for many to comprehend—that the contract does not cover the individual opinions of inspecting officers, but only an accurate and just interpretation of the specifications.

The Government, as well as the contractor, is responsible

for the action of its agents, and a matter definitely passed by an inspector stands as passed unless some special requirement permits of other action, or unless the circumstances are such that the contractor has clearly not delivered what was required.

Delay in Inspection.

This is a troublesome matter and at times difficult to obviate, when caused by reason of an overload of work on the inspection service. It is overcome by giving inspectors power and discretion in waiving inspection, accepting manufacturers' certified test reports, etc. For such delays as are beyond control of contractors, delay in delivery is allowed.

Coöperation between the manufacturers and inspection officers can obviate to a great extent delays in inspection.

Articles on which Inspection at Point of Manufacture was called for Shipped Without Inspection.

In such cases the article is often required to be shipped back at the expense of contractor. This is the contractor's or manufacturer's fault; but the constant recurrence of the practice sometimes leads a Government official to suspect that it is somtimes done to avoid inspection, especially when it is known that the article in question is rather urgently needed. The fact that an article is to be inspected at the place of manufacture is clearly stated in the contract, and shipment without inspection is generally inexcusable.

Delay in Receipt of Approved Drawings or in Obtaining Decisions on Questions of Design.

This may be the fault either of the contractor or of the Government. When the fault of contractor, it is usually due to the fact that drawings submitted are inadequate or inaccurate and not in sufficient detail to obtain a proper knowledge of the apparatus. Delay on the part of the Government in this matter is usually due to inadequate force to handle the matter or to lack of coöperation between different Government

forces or to indecision on the part of Government officials as to action to be taken, change of policy, etc.

Another practice sometimes indulged in is for contractors to supply a set of drawings that comply with requirements, but in actual manufacture to use another set or follow shop practices at variance with drawings. This may expedite things occasionally, but sooner or later will be shown up and all the advantage gained will be lost.

Representatives in Washington.

There is also in some quarters a feeling on the part of manufacturers that in order properly to do business for the Navy a Washington representative is necessary. While a representative at Washington is very desirable for the purpose of obtaining special and first-hand information, of being able better to supply information to the officials taking action in various matters and of explaining difficulties, such Washington representatives are not necessary, and a great amount of business is promptly and successfully done without them.

The idea that a Washington representative can influence the obtaining of business or obtain any special consideration in the waiving of deficiencies is a mistaken one. The best influence in obtaining business is a superior product for a lower price. Considerations as to waivers are given on a careful consideration of actual facts, and the Department is guided almost entirely by the recommendation of the local inspector concerned, who makes his recommendations from actual knowledge of the case in point.

DESCRIPTION AND TRIALS OF U. S. S. TORPEDO-BOAT DESTROYER *O'BRIEN*.

BY LIEUTENANT W. F. COCHRANE, U. S. N., MEMBER.

The *O'Brien* is one of six destroyers authorized by an Act of Congress approved August 22, 1912, these vessels being designated as Torpedo-Boat Destroyers Nos. 51 to 56, inclusive.

The contract for building three of these boats, Nos. 51, 52 and 53, named *O'Brien*, *Nicholson* and *Winslow*, was awarded to the Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa., and was signed December 7, 1912; the time allowed for completion being 23 months for the first boat, 23½ months for the second, and 24 months for the third.

The designed speed was 29 knots, at about 1,050 tons displacement.

The contract price for each vessel was $842,000.00, of which $502,000.00 was allotted for machinery.

The dimensions of vessel, machinery details, etc., are the same as those given in JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XXVII, No. 2, of May, 1915.

TRIALS.

The contract required—

(*a*) A progressive trial over the measured-mile course at Lewes, Del., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed

to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours' duration in the open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engines connected and in use.

(*e*) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engines connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engines.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engines connected and in use, at an average uniform speed of 12 knots, as nearly as possible.

(*g*) In addition to the above enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel oil and water consumption to be carefully measured on this trial.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 692.25 pounds at guaranteed maximum speed, 444.6 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

Standardization Trial (*a*).—This trial was first attempted on December 17, 1914, but was discontinued owing to cracking of hull plating. Data for this is not given.

The next trial was held February 8, 1915, with results in Table I-a.

The next was held April 8, 1915, with results given in Table I-b.

The last and successful trial was held May 6, 1915, with results given in Table I-c.

After this a two-hour trial at full power was made, the results of which are given in Table II, together with results on other two full-power trials, after the above mentioned standardization trials.

The balance of the *O'Brien's* trials were held in February, following the second standardization, and the data is not given, as the vessel was not retried at the lower speeds; the data given in Table IV is an average for all three vessels, taken from curves for the *O'Brien*.

Four-Hour, 12-Knot, Fuel-Oil and Water-Consumption Trial (*f*).—This trial commenced at 11:35 A. M., February 5, 1915, and was completed at 3:35 P. M., the same day. The weather was fair and the trial very successful. For data see Table II.

After the trial the *O'Brien* began the completion of her standardization trials.

Four-Hour Full-Speed Trial (*b*).—This trial began at 11:05 A. M., February 6, 1915, off Five Fathom Bank Lightship, and was completed at 3:05 the same day.

The weather was fair, except during the last of the run it was necessary to run head into the sea to avoid fog. The data obtained on this run is given in Table II.

Four-Hour, 24-Knot, Fuel-Oil and Water-Consumption Trial (*c*).—Following the full-speed trial, the 24-knot trial began at 3:25 P. M., and was completed at 7:25 P. M. The weather was fair, with rifts of fog. The trial was very successful.

Two-Hour, 15½-Knot, Fuel-Oil and Water-Consumption Trial (*g*).—The trial began at 5:10 P. M., February 7, 1915, and was completed at 7:10 P. M., February 7, 1915. The trial was very successful.

Four-Hour, 15½-Knot, Fuel-Oil and Water-Consumption Trial (d).—The trial began at 7:50 A. M., February 7, 1915, and was completed at 11:50 A. M., same day. The trial was very successful. The weather was fair.

Ten-Hour, 15½-Knot Endurance Trial (e).—This trial began at 12:30 A. M., February 8, 1915, and was completed at 11:05 A. M., the same day. The weather was fair. The trial was very successful, and the reciprocating engines and clutches operated excellently.

The trials of these three vessels are very interesting in view of the numerous trials conducted with different propellers and with the same propeller in slightly different positions. The *O'Brien* was the ship that made the majority of the various trials, and the performance of all three vessels was practically identical finally, although the *Nicholson's* oil consumptions were better than the *O'Brien's*, and the *Winslow's* considerably better than either of the other destroyers. This was due to vacuum and various other operating features and not to any difference in machinery.

The *O'Brien* began her first official standardization on December 17, 1914. This trial progressed favorably until the 29-knot group of runs began. On the second of three runs the after trimming tanks were reported full of water. This led to an examination. It was found that the hull plating in the wake of the propellers had cracked and the plates bent back. Photos 1 and 2 show the starboard and port sides of the vessel and the openings. The photos were taken after the trial before any repairs had been made. The plates were removed and the heavier plates fitted; stiffeners were also fitted. At the same time the struts were cut down slightly and made very much finer. After three alterations the *O'Brien*, on February 8, 1915, underwent a builders' official trial, the results of which are given in Table II and Curve No. 1. Plate I shows the speed and r.p.m. curve.

On this trial the speed was satisfactory, the oil consumptions were high, although under the guarantees, except at 29 knots. The vibrations of the vessel's hull over the propellers was

U. S. T. B. D. "O'BRIEN"—STARBOARD SIDE.

Forward end of break is vertical on frame 162, about 22 inches high, and extends aft 15 inches on top and 9 inches on bottom. The top is between the 9-foot and 10-foot water lines and the bottom is between the 7-foot and 8-foot water lines. The break apparently started with the vertical rupture at frame 162. The forward end of break is about in line with center of propeller. The distance from hull, at the break, to the propeller tip circle is about 18 inches.

U. S. T. B. D. "O'BRIEN"—PORT SIDE.

Vertical crack through plate on frame 162 at about same height as break on starboard side. The rag shown in the crack was drawn in as the water ran out in the dry dock.

excessive, and the vessel was not accepted. The problem confronting the contractors was an extremely difficult one, and also one about which as many theories as to cause, etc., can and were advanced as there are men to make them, but the only method open to the contractors was one of trial and retrial.

Accordingly a set of spare propellers for the *Aylwin* were obtained from the Government and placed on the U. S. S. *Nicholson*. The measurements of these propellers were as follows:

	Pitch.	Diameter.	Area.	Length of hub.
O'Brien	78 ins.	91½ ins.	3,996 ins.	24½ ins.
Nicholson	80 ins.	92½ ins.	4,062 ins.	22½ ins.

As can be seen, the propellers of the *Nicholson* were 1 inch greater diameter, 66 square inches larger area, 2 inches greater pitch, and the hub about 2 inches shorter. This brought the leading edge of the propeller 2 inches nearer the strut. With these propellers and some improvements in struts the *Nicholson* ran a very successful trial, the results of which were given in the JOURNAL, Volume XXVII, No. 2, of May, 1915. The vibration was not excessive, the oil consumptions were much better than the *O'Brien's*, and, above all, the *Nicholson* 563.4 r.p.m. for 29 knots, as against 611 r.p.m. for the *O'Brien* at the same speed. The cause of this performance was assigned to various reasons, among them was, difference in hull, + propellers, difference in struts, + propellers, and difference in struts + propellers + position of propellers. To ascertain the cause the contractors measured the hulls and found them the same to all intents and purposes, except the struts on the *Nicholson* were slightly finer.

The *O'Brien* was docked and a new set of propellers installed, having the same pitch, area and diameter as those of the *Nicholson*, but the hub was about 2 inches longer, this making the propeller fit without a distance piece and moving the leading edges of the propeller 2 inches farther away from the strut; also the hull was strengthened by the continuation

of an additional girder up to the main deck and across with full section instead of reduced section. With these alterations a trial was attempted. It was found the r.p.m was slightly less, also the vibrations, but the r.p.m for 29 knots were 593—again high.

The *O'Brien* was docked and the propellers removed and about 2 inches cut off the hub and face and the propellers replaced. This change placed the propellers in same position as the *Nicholson's*, the only difference being one of weight. With these propellers the *O'Brien* ran her final trial with the result that she required more r.p.m. than the *Nicholson*, the vibration was less than the *Nicholson* and the oil consumption on the 2-hour 29-knot run, the only run required, was far better than before.

Various reasons have been assigned as to the causes of this variation; the best seems to be, that the struts are not exactly in the stream lines, therefore causing broken water to the propellers, and the water close to the leaving edge of the strut is less broken up than a little farther back, and the moving of the propellers forward let them work in a much better area of water.

The performance of machinery in the vessels was par-excellent in all cases. The turbines have a minimum fore-and-aft clearance of $\frac{3}{16}$ inch, and a minimum tip clearance of $\frac{3}{32}$ inch, and are particularly fool proof, requiring no close adjustments, as in the dummy clearances of the Parsons turbines. The blading is extremely strong. It is far stronger than any other marine turbine blading in use at present.

In all respects the turbine is extremely well suited for one or both naval and merchant vessels, where maximum strength, combined with efficiency and ability to stand hard usage without continued watching of fine adjustments that require a very experienced man. The oil consumption of these vessels is higher than the Parsons turbine vessels at high speeds and is lower at low speeds, and is lower than one Parsons vessel at all speeds. Data given in Table IV shows the approximate data for various vessels.

The turbine is properly not a Zoelly turbine, but a Cramp turbine, and as turbines generally receive the name from the engineer of the plant in which they are developed, the credit for the development of this turbine belongs to Mr. J. F. Metten. There is hardly a feature in the turbine today that was in the original Zoelly turbines of the *Mayrant* and *Warrington*.

The tables give the following:

Table I (*a*), Standardization data second trial.
I (*b*), Standardization data third trial.
I (*c*), Standardization data fourth trial.
Table II, Data various trials made by *O'Brien* at same speed with different propellers.
Table III, Data for *O'Brien* and sister vessels at same speeds.
Table IV, Compares *O'Brien*, *Nicholson* and *Winslow* with *McDougal*, *Ericsson and Cushing*.
Plate No. I, Speed curves, Oil curves, S.H.P. curves — *O'Brien*.
Plate No. II, Curves *O'Brien* and sister vessels.

TABLE I (*a*).
STANDARDIZATION DATA U. S. S. *O'BRIEN*, LEWES, DEL., DECEMBER 14, 1914, AND FEBRUARY 5, 6 and 7, 1915.

No. of run.	Speed, knots.	Revolutions per minute.			Shaft horsepower.			Speed, Average group.	R.P.M. Average group.	S.H.P. Average group.
		Stbd.	Port.	Mean.	Stbd.	Port.	Mean.			
1	8.66	134.29	136.40	135.34	......	......	264			
2	8.40	148.22	146.96	147.59	......	......	295	8.51	143.63	304
3	8.59	142.79	145.21	144.00	......	......	360			
4	12.69	205.47	204.87	205.17	......	......	749			
5	11.56	204.90	204.69	204.79	......	......	820	12.27	204.64	794
6	13.27	204.48	203.12	203.80	......	......	785			
7	15.66	263.93	264.97	264.45	731	764	1,495			
8	15.68	263.69	263.34	263.52	796	785	1,581	15.81	263.82	1,547
9	16.22	263.61	263.92	263.77	783	748	1,531			
10	19.41	338.82	338.14	338.48	1,655	1,681	3,336			
11	20.63	339.09	339.17	339.13	1,656	1,669	3,325	20.04	338.82	3,327
12	19.49	338.64	338.46	338.55	1,654	1,666	3,320			
13	24.56	430.64	431.74	431.18	3,598	3,733	7,331			
14	23.27	431.18	432.76	431.97	3,756	4,023	7,779	23.94	431.10	7,596
15	24.64	427.24	431.30	429.27	3,636	3,859	7,495			
16	24.79	494.88	492.77	493.83	5,433	5,757	11,189			
17	27.09	494.29	494.56	494.43	5,351	5,704	11,055	25.93	494.15	11,146
18	24.76	498.72	497.11	497.92	5,475	5,808	11,283			
19	27.13	614.46	608.92	611.69	8,169	8,536	16,705			
20	30.46	605.18	598.28	601.73	8,149	8,076	16,224	28.66	600.16	16,037
21	26.59	589.57	581.42	585.50	7,332	7,660	14,993			
22	29.29	623.91	614.67	619.29	8,451	8,739	17,190			
23	29.41	627.39	615.76	621.58	8,340	8,724	17,064			
24	29.65	637.98	630.82	634.40	8,642	9,220	17,862	29.52	629.27	17,420
25	29.58	638.40	627.45	632.93	8,519	8,921	17,440			
26	29.53	638.99	635.01	637.00	8,527	8,871	17,398			

TABLE I (*b*).

STANDARDIZATION DATA U. S. S. *O'BRIEN*, LEWES, DEL., APRIL 8, 1915.

No. of run.	Speed, knots.	Revolutions per minute. Starboard.	Port.	Mean.	S.H.P., mean.	Speed, average group.	R.P.M. average group.	S.H.P. average group.
1	8.26	139.80	140.06	139.93	258			
2	9.85	139.98	141.39	140.69	246	8.82	140.19	245
3	7.31	139.08	139.77	139.43	230			
4	14.38	199.56	199.91	199.74	730			
5	10.66	200.97	200.21	200.59	742	12.51	199.98	738
6	14.35	198.66	199.33	199.00	737			
7	14.13	258.07	257.32	257.70	1,535			
8	17.84	257.66	257.88	257.77	1,600	15.99	257.40	1,569
9	14.16	256.18	256.85	256.37	1,540			
10	22.51	337.20	338.80	338.00	3,603			
11	19.01	337.91	338.57	338.24	3,571	20.78	338.64	3,610
12	22.58	339.77	340.34	340.06	3,694			
13	22.63	422.71	422.09	422.40	7,697			
14	22.68	419.95	421.77	420.86	7,667	24.21	421.64	7,644
15	22.89	421.74	423.11	422.43	7,546			
16	27.21	479.29	479.51	479.40	10,871			
17	24.97	479.64	481.14	480.39	10,964	25.97	478.55	10,884
18	26.73	472.48	475.54	474.01	10,735			
19	27.91	586.49	589.32	587.91	16,096			
20	29.46	584.54	586.60	585.57	16,209	28.81	587.11	16,181
21	28.41	588.93	589.85	589.39	16,209			
22	30.13	613.61	615.17	614.39	17,830			
23	29.58	621.46	620.55	621.01	18,126			
24	30.05	619.20	621.80	620.50	17,982	29.92	621.39	18,091
25	30.03	625.08	625.76	625.42	18,096			
26	29.85	622.28	623.37	622.83	18,487			

TABLE I (*c*).

STANDARDIZATION DATA U. S. S. *O'BRIEN*, LEWES, DEL., MAY 6, 1915.

No. of run.	Speed, knots.	Revolutions per minute.			S.H.P., mean.	Speed, average group.	Revolutions per minute, ave. group.	Shaft horse-power, ave. group.
		Starboard.	Port.	Mean.				
1	15.14	257.14	255.20	256.17	1,514			
2	16.45	256.00	253.48	254.74	1,479	15.60	254.77	1,473
3	14.34	253.57	253.24	253.41	1,421			
4	21.82	336.72	334.62	335.67	3,377			
5	19.04	339.28	338.27	338.78	3,391	20.45	336.89	3,372
6	21.91	334.04	334.57	334.31	3,329			
7	22.87	430.45	431.41	430.93	7,592			
8	25.97	424.89	425.61	425.25	7,427	24.39	427.14	7,498
9	22.74	427.76	426.46	427.11	7,547			
10	28.06	485.31	484.80	485.06	11,022			
11	24.76	484.98	482.59	483.79	10,728	26.41	484.64	10,899
12	28.06	486.88	484.96	485.92	11,116			
13	27.50	574.11	572.69	573.40	15,011			
14	30.51	575.26	572.08	573.67	15,623	29.03	574.62	16,469
15	27.59	578.54	576.94	577.74	15,618			
16	30.80	586.97	587.82	587.40	16,257			
17	28.37	604.37	604.02	604.20	17,086			
18	30.93	596.99	597.37	597.18	16,737	29.68	599.47	16,751
19	28.48	598.41	600.24	599.33	16,377			
20	31.01	607.16	606.62	606.89	17,346			

TABLE II.—U. S. S. "O'BRIEN;" FOUR-HOUR FULL POWER.

	4 Hours.	1 Hour.	2 Hours.
Date	2/6/15	4/8/15	5/7/15
Displacement	1,052.0	1,053.0	1,053.7
Boilers	4.0	4.0	4.0
Heating surface	21,600.0	21,600.0	21,600.0
Speed, in knots	29.168	29.03	29.053
R.P.M., starboard	618.31	593.49	575.78
port	615.19	593.83	574.97
mean	616.75	593.66	575.38
S.H.P., mean	16,275.0	16,620.0	15,386 0
Pressures:			
Main steam	245.0	251.0	249.0
Full speed (abs.)	215.0	211.0	206.0
Cruising speed (abs.)	170.0	160.0	150.5
Tenth stage (abs.)	28.0	26.5	25.2
Fourteenth stage (abs.)	16.5	15.3	15.7
Gland steam	6.4	5.0	5.0
Auxiliary exhaust	7.2	7.0	6.0
Vacuum, starboard	27.0	29.0	28.5
port	27.5	29.2	28.7
Lubricating oil to M.E.	11.4	11.0	10.1
Fuel oil per knot run	794.24	707.32	643.93
Propeller, pitch, inches	78.0	80.0	80.0
diameter, inches	91½	92½	92½

TABLE III.—FOUR-HOUR, FULL POWER.

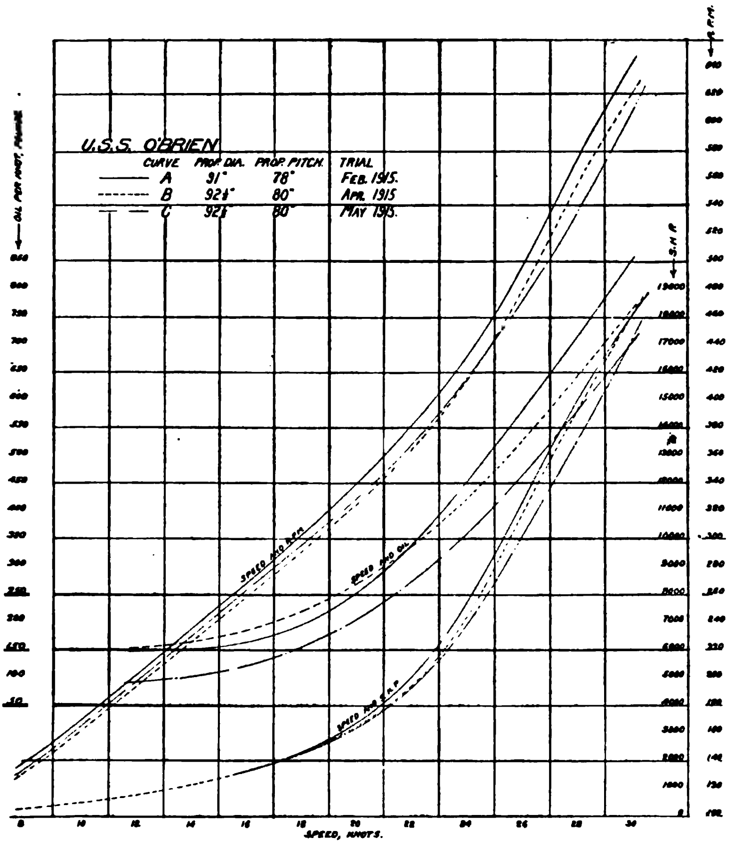
	O'Brien.	*Nicholson.*	*Winslow.*	*Balch.*
Date of trial	5/7/15	3/24/15	6/30/15	2/26/14
Displacement	1,054.7	1,045.0	1,041.0	1,053.9
Boilers in use	4.0	4.0	4.0	4.0
Heating surface	21,600.0	21,600.0	21,600.0	21,600.0
Speed in knots	29.053	29.084	29.054	29.618
R.P.M., starboard	575.78	566.40	573.21	596.96
port	574.97	566.79	573.09	597.17
mean	575.38	566.10	573.15	597.06
S.H.P., mean	15,386.0	15,906.0	15,984.0	17,251.0
Pressures:				
Main steam (G)	249.0	250.8	250.0	248.0
Full speed (abs.)	206.0	208.2	207.0	215.0
Cruising speed (abs.)	150.5	157.5	152.5	200.0
Tenth stage (abs)	25.2	25.2	25.3	31.6
Fourteenth stage (abs.)	15.7	14.3	15.0	......
Gland steam (G)	5.0	6.3	6.0	8.3
Auxiliary exhaust (G)	6.0	8.0	8.0	6.0
Vacuum, starboard	28.5	27.6	28.0	28.9
port	28.7	27.8	27.8	29.03
Lubricating oil to M.E.	10.1	12.6	10.0	15.7
Fuel oil per knot run at 29 knots	643.93	687.3	661.50	753.54

Propellers same for all vessels.

TABLE IV.—FULL POWER.

	O'Brien, Nicholson, Winslow. Cramps' turbines.	*McDougal.* Parsons turbines.	*Ericsson.* Parsons turbines.
Displacement, tons	1,045.0	1,021.0	1,087.1
Speed	29.06	30.697	29.29
R.P.M., average	571.0	608.99	616.02
S.H.P., average	15,758	16,974	17,479
Vacuum	......	......	......
Fuel oil per knot run at speed given above	649.5	645.0	746.2
Twenty-four Knots.			
Displacement, tons	1,046.0	1,013.4	1,047.8
Speed	24.081	24.074	24.115
R.P.M.	416.0	417.27	433.32
S.H.P.	7,300	6,413	7,150
Vacuum	......	......	......
Fuel oil per knot run at 24 knots, water	388.3	325.0	392.55
Fifteen and one-half Knots.			
Displacement, tons	1,053.0	1,019.0	......
Speed	15.59	15.55	15.448
R.P.M	251.0	256.45	258.04
S.H.P	1,600	1,236	1,099
Vacuum	......	......	......
Fuel oil per knot run at 15½ knots,	138.0	148.0	193.2
Twelve Knots.			
Displacement, tons	1,050.0	1,020.0	1,096.3
Speed	12.08	12.51	12.056
R.P.M	194.2	195.45	198.62
S.H.P	672	517	435
Vacuum	......	......	......
Fuel oil per knot run at 12 knots.	105.0	129.6	156.3

PLATE I.


U.S.S. O'BRIEN
CURVE PROP. DIA. PROP. PITCH TRIAL
A 91" 78" FEB. 1915.
B 92½" 80" APR. 1915
C 92½" 80" MAY 1915.
SPEED AND R.P.M.
SPEED AND OIL
SPEED AND S.H.P.
OIL PER KNOT, POUNDS
S.H.P.
R.P.M.
SPEED, KNOTS.

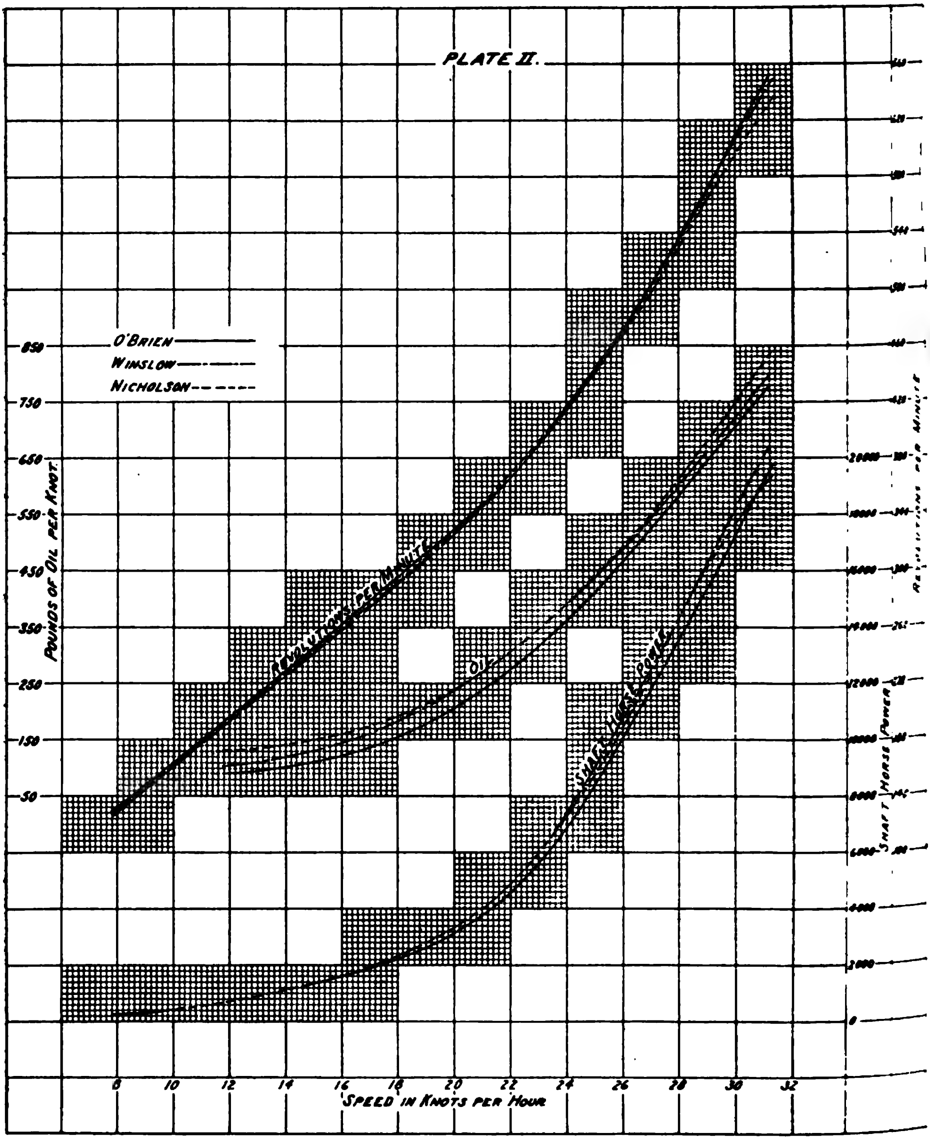


PLATE II.
O'BRIEN
WINSLOW
NICHOLSON
POUNDS OF OIL PER KNOT.
850
750
650
550
450
350
250
150
50
REVOLUTIONS PER MINUTE
OIL
SHAFT HORSE POWER
8
10
12
14
16
18
20
22
24
26
28
30
32
SPEED IN KNOTS PER HOUR
SHAFT HORSE POWER
REVOLUTIONS PER MINUTE
0
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000

NOTES ON STORAGE BATTERIES.

By Lieutenant C. S. McDowell, U. S. N., Member.

In the following notes no attempt has been made to treat any one of the various subjects under storage batteries thoroughly, or to go deeply into the theory of the storage battery or accumulators, as there are certain standard books available, such as: "Storage Battery Engineering," by L. Lyndon; "Theory of the Lead Accumulator," by F. Dolezalib; and various handbooks and trade catalogues of the battery manufacturers. Some of the questions affecting design and operation are presented with the idea of encouraging a further study of the subject by officers and others who have to do with the use of storage batteries in the Navy.

A storage battery is a vehicle for the chemically storing up of energy. Electricity itself cannot be stored, so that if it is necessary to use it at some time other than when generated it is necessary to convert it to some other form of energy and then reconvert it back to electricity when desired.

Storage batteries were originally used to carry the peak load, smooth out the load factor, and act as a stand-by source in central stations; lately, however, the combining and interlinking of the various large systems and the introduction of alternating current has removed to a great extent the necessity for their use in this capacity at all but isolated plants. Of late years the principal output of the battery manufacturers has been for electric vehicles, train lighting, gas car starting and lighting and submarines. In all the above uses a fairly rugged battery of minimum weight and space is required, so that the later developments in storage-battery engineering have been along those lines, which is fortunate for the Navy, for

we must depend for improvements and developments along the Navy's particular lines upon the commercial development along similar lines.

TYPES.

There are at present two widely different types of storage batteries, the lead acid and alkaline. Each type has certain advantages and certain disadvantages; for certain services one type is preëminent and for other purposes the other is best. Each particular class of storage-battery installation should be carefully studied and worked out to obtain the desired results, and both types of batteries should have their inherent characteristics weighed to bring out the advantages and disadvantages for the particular service to which the installation will be put. Neither type of battery is a cure-all for all evils; thus for central-station stand-by service where a large battery is installed and where there may not be more than three or four cycles a year of charge and discharge, some form of lead battery would naturally be installed on account of cost alone, and in general for stand-by service the lead-paste type would be chosen. On the other hand, where a battery for any service may be expected to have very hard mechanical usage, where it will be discharged practically every day and where it will likely be left standing idle while discharged and in general not receive much attention, such as batteries installed in dories and other small boats for lighting, etc., the nickel-iron alkaline (Edison type) would naturally be chosen over lead type, now in common use, on account of its much greater mechanical strength and ability to stand such service for a long time. In this case the cost per year should be less than with the lead types. Certain batteries are installed for "floating on the line," that is, connected in parallel with the normal power supply, so that when the regular supply fails the battery immediately takes up the load. Such an installation acts as the air chamber on a pump, in that if supply voltage rises, the battery will start charging, taking power from the line, while if the supply voltage falls, the battery will discharge to the

line, in each case tending to keep the line voltage constant. When floating on the line the battery should be in a charged condition, and the normal voltage of the supply should be just at the closed-circuit voltage of the battery and high enough so that the battery, if discharged, will take practically a full charge. In addition to the above the discharge-voltage curve of the battery should be flat enough to keep within the voltage limits necessary for the apparatus served by the line. Figures 1 and 3 show characteristic charge and discharge curves of the Edison type of cell, and Figures 2 and 4 show characteristic charge and discharge curves of a lead cell.

It will be seen that the range in voltage on an Edison cell is from 1 volt discharged to 1.8 end of charge, while on a lead cell it is from 1.8 to 2.65 volts at normal rates. The Edison battery will float on the line and take a partial charge at 1.55 volts per cell, while the lead storage battery will float on the line and charge, if discharged, at 2.3 volts per cell. Thus on what is termed 20-volt service, with the lead battery the generator should give 23 volts (10 lead cells used), and if the generator fails the battery will pick up the load at 20 volts, and the drop will be gradual down to 18 volts, and when the main power supply is cut in again the battery will practically pick up a full charge. Under the same conditions 16 Edison cells might be used, in which case the battery could float on the line with the generator giving 24.8 volts; if the generator failed the battery would maintain the supply from 22.5 volts (approximately) to 16 volts, and when the generator was again on the line the battery would take only a partial charge, the battery would have to be charged separately or the voltage raised. The range in voltage of the Edison battery on discharge would also be greater than the limits required by the usual 20-volt instruments.

The above cases are only given to emphasize the statement that because one type of battery is best for one service is no reason for assuming that it will be best for an entirely different class of service.

ELECTROLYTE.

For any type of battery the question of purity of electrolyte is of the utmost importance. In the Edison battery the electrolyte consists of a 21 per cent. solution of potassium hydrate in pure distilled water with a small percentage of lithium hydrate. To insure pure electrolyte being used the Edison Company require in their guarantee that only electrolyte as furnished by them be used. The Navy purchases this in the powdered form to save the transportation of useless weight of water, but when mixing and when adding water to make up for evaporation, only pure distilled water should be used. The amount of water required for make up in the Edison cells is, as a rule, greater than in the lead, due to the greater electrical losses when charging and thus greater generation of heat and evaporation of water. In lead-cell installations with individual cell ventilation, however, excessive evaporation is obtained unless the blowers are carefully regulated. Even with the use of pure distilled water in the Edison cell the electrolyte becomes gradually converted to potassium carbonate through absorbtion of carbon dioxide, with a consequent reduction of capacity of the cell, so that after about 250 cycles this electrolyte should be renewed. The distilled water used should be freshly distilled and not allowed to become aerated.

In the electrolyte for lead batteries the allowable impurities in the acid are specified, and should not be exceeded. Acid procured in accordance with Navy standard specifications is always well within the allowable limits, and the impurities found in electrolyte in cells which have had considerable use can seldom be traced to the acid. The greatest opportunity for impurities to get in the electrolyte is through the water added to make up for evaporation. As the impurities do not evaporate with the water, very small quantities of iron in the water will cumulate and eventually increase to such a proportion as to be harmful, causing local action, the giving off of hydrogen when standing idle, and the reduction in capacity of the cells. There is a certain amount of impurities in the

lead plates, both positive and negative, a chemical analysis of a number of plates of the paste type shows approximately .06 per cent. of iron present. The impurities in the plates will be gradually absorbed by the electrolyte, and as the total iron in the electrolyte should be kept lower than .01 per cent., it is seen that the iron in the plates may be very injurious. This iron probably gets in to the paste due to the use of iron rollers in grinding the active material, and great care by the manufacturers is required to overcome this. Another probable source of impurities is from the rubber jars and separators, both of which contain a certain percentage of iron. However, as stated previously, the greatest source of danger from impurities is in the added distilled water. The absorptive power of distilled water is second only to that of acids, all but the precious metals and block tin are suspectible to its action, and if it comes in contact with tanks or pipes or other metals, or wood, it will be found to contain slight quantities of them in solution.

The specific gravity of the electrolyte is stated as at a certain temperature and with the cell charged, the Navy have adopted 80 degrees F. as the standard reference temperature. The rate of diffusion of acid into the pores of the plates increases with the temperature, so that the capacity of a cell is markedly increased at higher temperatures.

The recommended gravity of the electrolyte varies with the types of plates, the amount of separation, etc., and it is usually stated by the manufacturer. In Figure 6 curves are given to show the increase in capacity when using higher-gravity electrolyte; 1.250 is usually, however, taken as the maximum, for although the capacity is increased when using stronger electrolyte, the life of the cell is materially shortened, and this should be resorted to only in emergencies when it may be desired to get more than rated capacity out of the cell. The effect of impurities is greatly increased in concentrated electrolyte.

In the lead battery the state of charge may be accurately

obtained from the gravity of the electrolyte provided the gravity when fully charged and at a certain temperature is known. In addition to determining that a battery is fully charged by means of the gravity, it is usual to keep charging until the voltage has remained constant and the battery is gasing freely. In the Edison battery, however, the state of charge cannot be accurately obtained from the gravity on account of the fact that the specific gravity of the displaced material is approximately the same as the electrolyte itself.

CHARGING.

In Figures 1, 2, 3 and 4 two methods of boosting are shown for charging to near full capacity in a minimum of time. Overcharge at high rates is injurious, but the only factors in high-rate charging which act injuriously are gasing and heating, and these occur only when more current is being passed through the battery than the plates can store. Therefore any current rate which the cells will absorb without gasing is not injurious, and it is upon this principle that boosting is applied. The nearer discharged the battery the higher charging current it can take, and by starting the charge at a high rate and tapering to a low rate a large proportion of the discharge can be put back in very short time. Where conditions permit, charging at a constant potential is probably the ideal method since it is automatic and requires little attention. In the case of a lead battery it requires 2.3 volts per cell of the battery, and in the case of the Edison battery approximately 1.7 volts per cell of the battery. In Figure 2 there is shown characteristic curves of constant potential charging of a lead submarine cell. In this case a charge of approximately 80 per cent. full capacity has been put into the battery in two hours and fifteen minutes, and approximately 50 per cent. of its full capacity has been put in within one hour. The curves in Figure 1 show an Edison cell charging in series with the lead cell, the current being the same in both cases but the voltage curve of the Edison cell is not a straight line due to the different in-

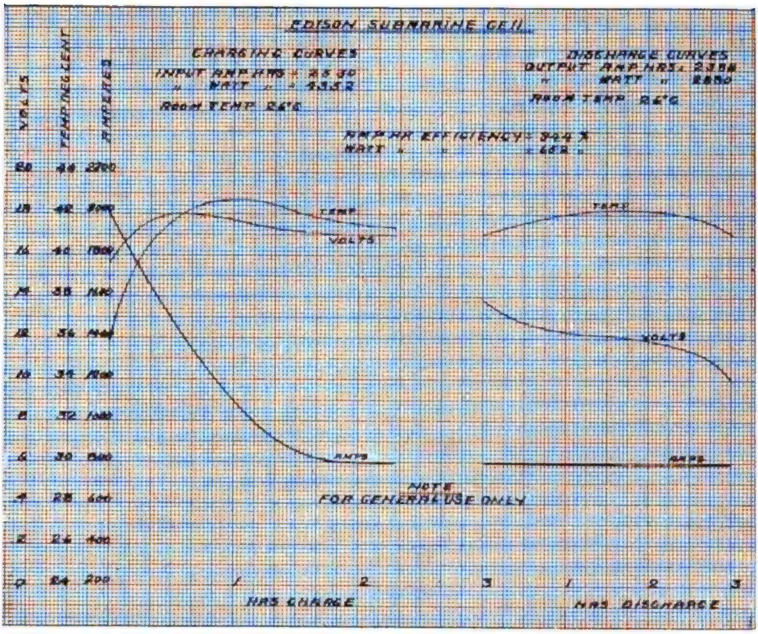

EDISON SUBMARINE CELL
CHARGING CURVES
DISCHARGE CURVES
VOLTS
TEMP DEG CENT
AMPERES
TEMP
VOLTS
AMPS
NOTE
FOR GENERAL USE ONLY
HRS CHARGE
HRS DISCHARGE

FIG. 1.

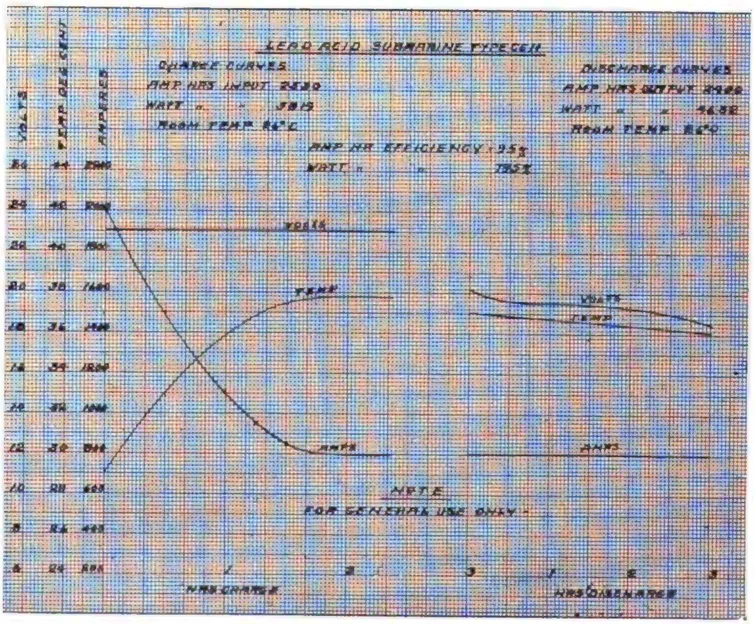

LEAD ACID SUBMARINE TYPE CELL
CHARGE CURVES
DISCHARGE CURVES
VOLTS
TEMP DEG CENT
AMPERES
VOLTS
TEMP
AMPS
NOTE
FOR GENERAL USE ONLY
HRS CHARGE
HRS DISCHARGE

FIG. 2.

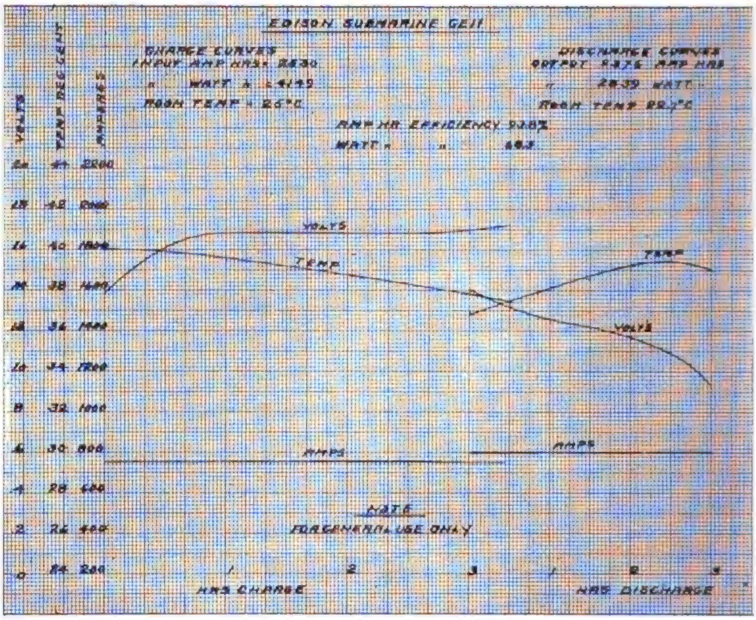

EDISON SUBMARINE CELL
CHARGE CURVES
DISCHARGE CURVES
AMP HR EFFICIENCY
VOLTS
TEMP DEG CENT
AMPERES
VOLTS
TEMP
AMPS
NOTE
FOR GENERAL USE ONLY
HRS CHARGE
HRS DISCHARGE

FIG. 3.

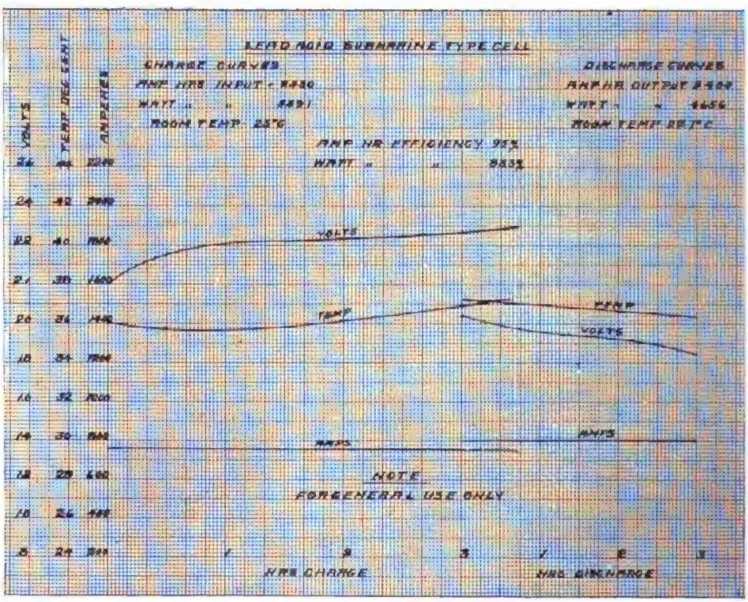

LEAD ACID SUBMARINE TYPE CELL
CHARGE CURVES
DISCHARGE CURVES
AMP HR EFFICIENCY
VOLTS
TEMP DEG CENT
AMPERES
VOLTS
TEMP
AMPS
NOTE
FOR GENERAL USE ONLY
HRS CHARGE
HRS DISCHARGE

FIG. 4.

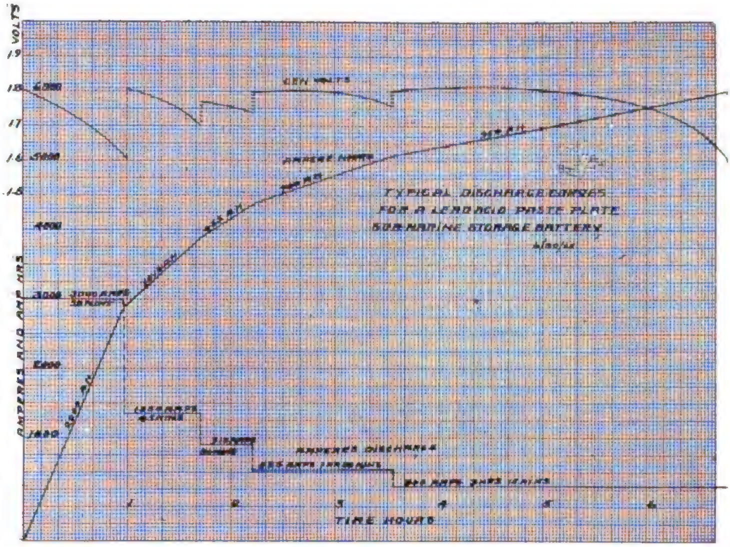

TYPICAL DISCHARGE CURVES
FOR A LEAD ACID PASTE PLATE
SUBMARINE STORAGE BATTERY
VOLTS
TIME HOURS

FIG. 5.

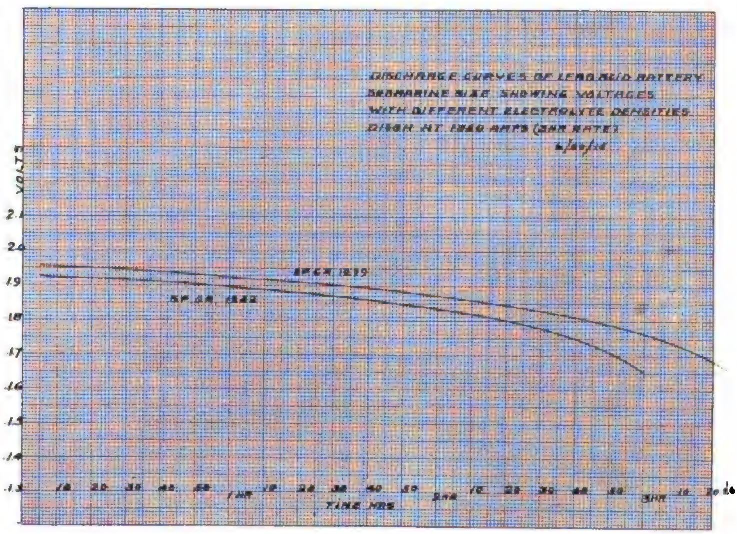

DISCHARGE CURVES OF LEAD ACID BATTERY
SUBMARINE SIZE SHOWING VOLTAGES
WITH DIFFERENT ELECTROLYTE DENSITIES
VOLTS
TIME HRS

FIG. 6.

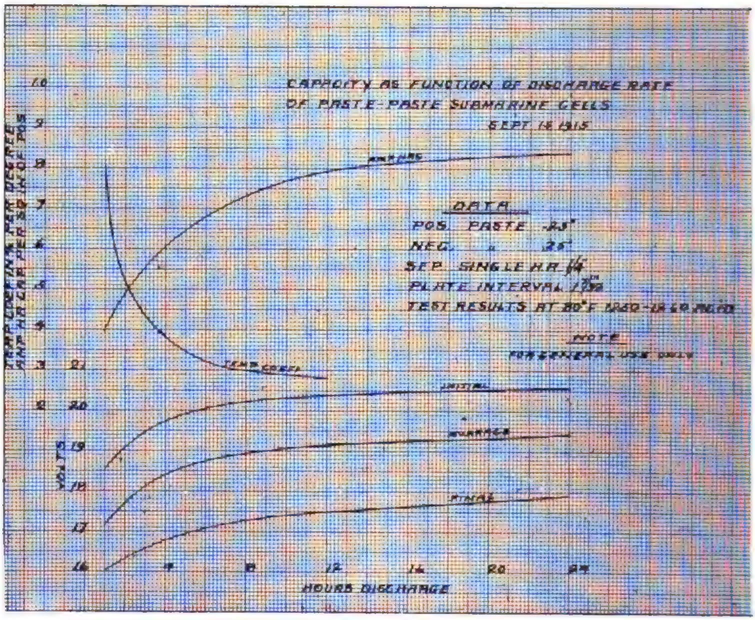

CAPACITY AS FUNCTION OF DISCHARGE RATE
OF PASTE-PASTE SUBMARINE CELLS
SEPT 18 1915
DATA
POS. PASTE
NEC.
SEP. SINGLE H.R.
PLATE INTERVAL
TEST RESULTS AT 80°F
NOTE
FOR GENERAL USE ONLY
TEMP. COEFF.
INITIAL
AVERAGE
FINAL
VOLTS
HOURS DISCHARGE

FIG. 7.

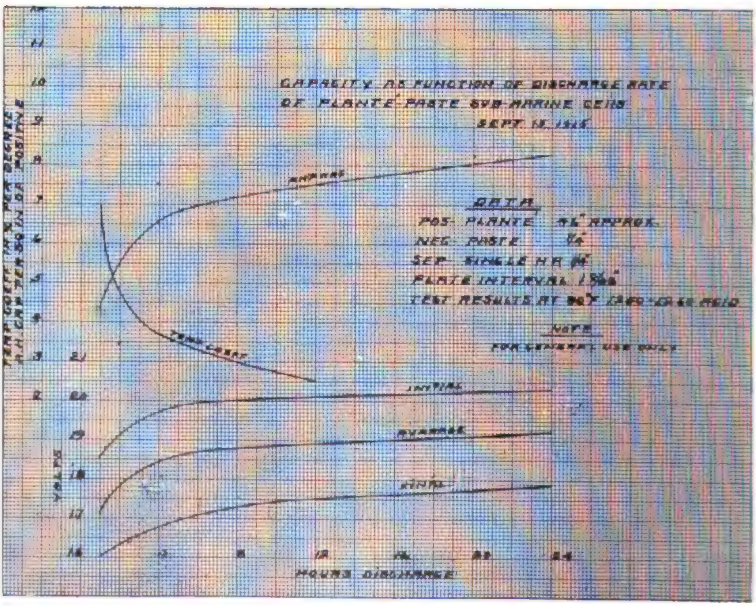

CAPACITY AS FUNCTION OF DISCHARGE RATE
OF PLANTE-PASTE SUB-MARINE CELLS
DATA
POS. PLANTE
NEG. PASTE
SEP. SINGLE H.R.
PLATE INTERVAL
NOTE
FOR GENERAL USE ONLY
TEMP. COEFF.
INITIAL
AVERAGE
FINAL
VOLTS
HOURS DISCHARGE

FIG. 8.

ternal characteristics. The limiting temperatures in charging for the lead battery are approximately 44 degrees C. (110 degrees F.), and for the Edison 46 degrees C. (115 degrees F.). It will be noted that in neither case shown by the figures does the temperature approach the above limits.

In the Figures 1 and 2 there are also given the discharge curves of the two types of cells at the 800 ampère rate, it will be noted that the ampère-hour efficiency and watt-hour efficiency in both types is higher than that obtained when the cells are fully charged. In addition to the advantages already given, a battery would apparently have a longer life when worked from 10 per cent. charged to 90 per cent. charged than when worked to the limits in both directions. It is necessary, however, to give periodic overcharges to keep the battery up to full capacity. The principal deterioration of a cell is the shedding of active material which is caused by the gasing and large variations in temperature, and in boosting both of these are minimized.

In certain cases it is more convenient to boost at a constant current on account of the limits of the generator and the high initial ampèrage required by the constant voltage method. To determine what current rate can be safely used for boosting, the following rule may be used:

$$\text{Charging current in ampères} = \frac{\text{Ampère hours discharged.}}{\text{1 plus hours available for charge.}}$$

As an example, if 2,400 ampère hours have been taken from the battery, and one hour is available for charging.

$$\text{Charging current } \frac{2,400}{1 \text{ plus } 1} = 1,200 \text{ ampères.}$$

The above method will not put in as great a charge in the same time as the constant-potential method, and the current must not be continued beyond the time for which the rate is figured, or gasing and heating will result.

In Figure 5 is shown by curves the recovery of a lead bat-

tery after discharge at high rates. When discharging at the one-hour rate the battery shows low efficiency due principally to the fact that all the energy stored cannot be taken at such a rate on account of the poverty of the electrolyte below the surface of the active material, and the discharge is mostly the surface capacity only. After shifting to a lower rate the acid diffuses in to the inner mass of the active material, and by gradually changing the rate of discharge a total capacity equal to the 20-hour rate is approximately obtained as shown. The 20-hour rate of this particular cell shown is 300 ampères, or a total of 6,000 ampère hours. It should be noted that this total ampère hours has been taken out in six hours and fifty minutes. Discharging in this manner is practically the reversal of the "boosting" method of charging.

In Figures 7 and 8 the effect of the temperature on capacity of paste positive, paste negative lead cells, and Plante positive, paste negative lead cells is shown by curves, and in addition curves are given which show the effect on capacity of the rate of discharge. These curves show the general characteristics only, and should not be applied to any particular case, as the results vary with the material and design of plates, with the method and the amount of separation, and strength of electrolyte.

In order to obtain the full capacity from a battery both the internal resistance and the external resistance should be kept at a minimum, the internal resistance is, in general, controlled by the design, but the external resistance can generally be controlled and kept low by properly looking out for the connections to see that good electrical contact of sufficient area is obtained. It is extremely difficult to keep a good electrical contact when lead connections are bolted together, because lead oxide tends to form, which acts as an insulator; for this reason it is desirable to lead burn all connections wherever possible. As an illustration of the difficulties experienced in bolted connections the case of certain cells may be taken in which the plates of the cell are divided in three groups, the

different groups being connected by bolted equalizing bars. It has been noted that, in a number of cases of such construction, the drop across the connections has been as great as 50 millivolts instead of approximately 2 millivolts which should be obtained. The effect is to throw most of the work on one or two of the groups which, when connected in series with cells of full capacity, will be overcharged and overdischarged, and in cases reversed, giving rise to buckled and cracked plates and short circuits. Some cases have been noted where one or two of the groups of a cell were completely insulated, causing the full capacity to be taken from one or two groups.

SEPARATORS.

The purpose of the separators is to keep the plates from short circuiting by mechanical contact, to keep the separation between plates sufficient so that the required amount of electrolyte may be present between them, to be of sufficient porosity for the diffusion of the electrolyte, and should contain no materials which will injuriously affect the plates. Considerable trouble is experienced in obtaining separators which meet all the requirements. Those principally used are single hard rubber, double rubber, wood, and wood and rubber. In the case of wood separators it is very difficult to get them entirely free of acetic acid and, as a rule, they should not be used against the positive plates; the effect on the negative plates is not so harmful, and when used in conjunction with hard rubber the wood separators should be next to the negative plates. Wooden separators when not in use should be kept submerged in distilled water so as to prevent their drying out and becoming brittle. In the case of single-rubber separators a considerable number of holes are necessary to give sufficient porosity, these holes provide a path for the lead trees found in the electrolyte to bridge the space between plates and cause short circuits; in addition, it is difficult to obtain a design of single-rubber separator which will remain in a plane—

the separators tend to take on a waved surface, forming pockets in which the shedding active material collects and short circuits the plates. Short-circuited plates should be readily detected by the fact that the cells in which they are contained cannot be kept up to capacity and voltage. The best practice at present is to install either double-rubber or wood-and-rubber separators.

DESCRIPTION AND TRIALS OF U. S. S. *FULTON.*

(*Submarine Tender No. 1.*)

BY LIEUTENANT (J. G.) C. N. HINKAMP, U. S. N., MEMBER.

The builders, Fore River Shipbuilding Company, and the contractors, The New London Ship and Engine Company, have recently successfully completed the final trials of the U. S. S. *Fulton.*

Officially known as *Submarine Tender No. 1*, the *Fulton* is the first submarine tender built for the U. S. Navy, and was authorized by an act of Congress, approved March 4, 1911. The contract for her construction was signed June 19, 1912, the time of completion being twenty-four months and the price $492,930.00. She is a single-screw vessel, driven by Diesel engines, connected to the propeller shaft by a jaw clutch, and designed for a speed of 12.25 knots at a displacement of 1,407 tons, with the engines developing about 1,000 horsepower.

PRINCIPAL HULL DIMENSIONS.

Length on L.W.L., feet and inches, . . .	216–00
Length, over all, feet and inches, ...	226–06
Beam on L. W. L., feet and inches, ...	35–01.3125
Beam, extreme, feet and inches, ...	35–01.3125
Draught to L.W.L., mean, feet and inches, .	12–11.5
Displacement corresponding, tons, ...	1,407.00
tons per inch immersion at L.W.L, ...	12.36
Area immersed, midship section, square feet, .	377.46
L.W.L. plane, square feet, .	5,200.00
Coefficient of fineness, block, ...	.533
midship section, . . .	.831
L.W.L., ...	.808

GENERAL DESCRIPTION OF HULL.

The vessel is of the flush-deck type. There are two masts, fitted with wireless, signal yards, etc., and one smoke pipe.

Main Deck.—The main deck is continuous from stem to stern. On it are located the windlass, torpedo tubes, pilot house, deck house, towing engine and electrolyte tank. The deck house contains the flotilla commander's and commanding officer's quarters, the ship's and submarines' offices, galleys and sick bay. On top of the deck house there is an electric winch used for hoisting boats. There is also a steering platform and an acid tank, besides the boat stowage. On top of the pilot house and extending to the sides of the vessel is the bridge, fitted with steering wheel, searchlight and other navigational instruments.

Second Deck.—This deck extends from the bow to the engine space forward, and aft from the engine space to the stern. Foward are located the officers' storerooms and quarters, wardroom and machine shop. Aft are located the petty officers' and crew's quarters.

Platform Deck.—This deck extends from the bow to the boiler room, and from the engine space to the stern. Forward are located the general storeroom, submarine storerooms, torpedo stowage, refrigerator rooms, armory, and trimming tanks. Aft are crew's quarters.

Hold.—In the hold, forward and aft, are located trimming tanks, fuel-oil tanks, fresh-water tanks, magazines and stores. The chain lockers are forward, and the machinery spaces occupy the entire midportion of the hold. The engineers' storerooms are located in the shaft alley.

Battery.—The battery consists of a single twin-deck torpedo tube.

Boats.—

2 35-foot motor boats.
2 33-foot motor sailing launches.
2 21-foot motor dories.
3 17-foot pulling dories.
2 28-foot whale boats.
1 12-foot punt.

Complement.—

1 Division commander (submarine).
1 Commanding officer (tender).
14 Wardroom officers (submarine and tender).
174 Petty officers and crew (submarine and tender).
190 Total for one division.

Fire Main.—The fire main, size 2.5 inches, is supplied by one fire and bilge pump. Branches for the fire plugs are taken off at convenient locations. The eight fire plugs are located as follows:

No. 1, forecastle, main deck.
No. 2, starboard side amidships, main deck.
No. 3, port side amidships, main deck.
No. 4, aft of deck house, main deck.
No. 5, wardroom country, second deck.
No. 6, compartment C-6, near forge, second deck.
No. 7, compartment D-9, crew space, second deck.
No. 8, compartment D-5, crew space, platform deck.

Fresh Water.—Fresh water for drinking and washing purposes is stowed in tanks in the hold in compartment A-5, capacity 24 tons. Reserve feed water to the extent of thirteen tons is carried in addition to the supply for the ship's use.

Drainage System.—A 4-inch drain extends the entire length of the vessel with connections to the fire and bilge pump, also to all compartments and trimming tanks. Macomb strainers are fitted near the pump suction.

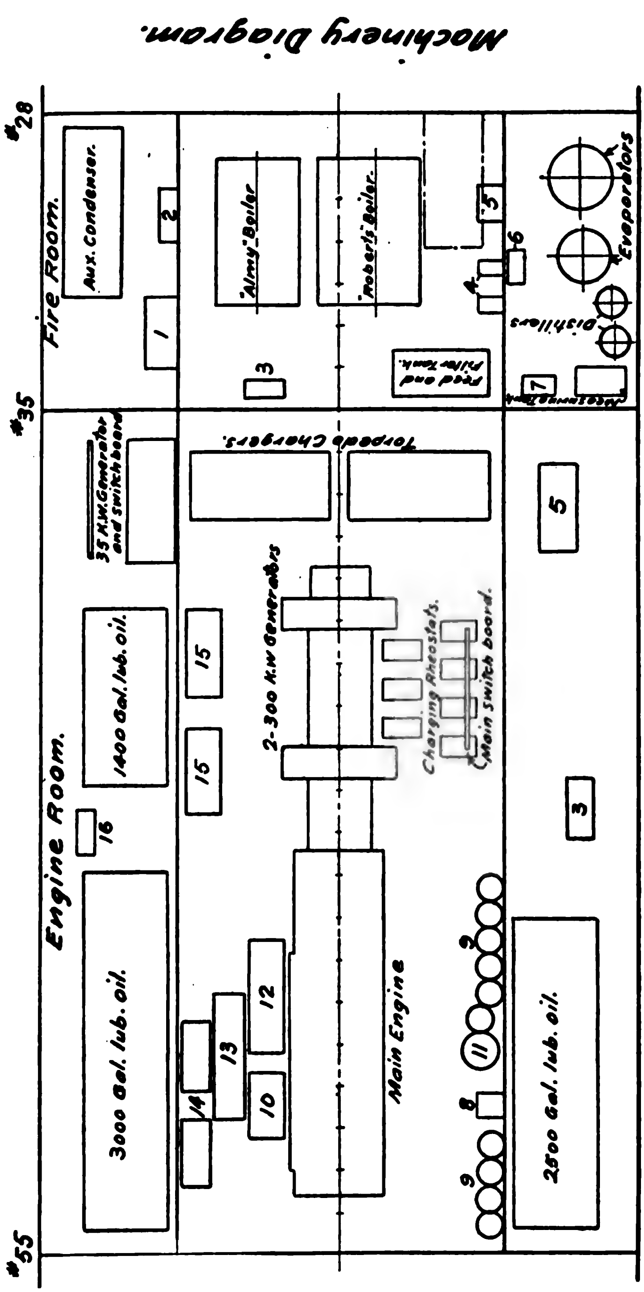
Ventilation.—Forced ventilation is used for the living spaces. There are two circuits, one forward and one aft. The forward blower is located in compartment B-5, the after blower in compartment D-9.

Heating.—The vessel is heated by steam, the steam being supplied by the auxiliary boilers. Radiators are fitted where necessary.

DESCRIPTION OF MAIN ENGINES.

The arrangement of machinery is shown in Plate I.

The main engine consists of a single, vertical, inverted, two-cycle, single-acting, air-starting-and-reversing oil engine



1 Distiller circ. pump.
2 Sanitary pump.
3 Fuel oil feed pump.
4 Main & Aux. feed pump.
5 Fire and bilge pump.
6 Evaporator feed pump.
7 Dist. fresh water pump
8 Westinghouse Compressor.
9 Engine air flask.
10 Air reservoir.
11 Water Cooler.
12 Oil Cooler.
13 Fresh water tank.
14 Air accumulators.
15 Circ. W. & oil pump.
16 Lub. oil pump.

PLATE I.

of the Diesel type, with six working cylinders and two air compressors. The shaft horsepower is about 1,000 at 260 r.p.m. The six working cylinders and the air-compressor cylinders are are on the same fore-and-aft line, the air compressors being at the forward end of the engine, where the control station is located. The starting, stopping, reversing and changing of speed are all controlled by a hand wheel at this station. The pistons are of the stepped type, the upper piston being the working piston, the lower piston being the scavenger piston. From the scavenging cylinders the scavenging air at low pressure is carried to an external receiver whence it goes to the working cylinders, and overboard through the exhaust. The crank pit is entirely enclosed. Each working cylinder is provided with two scavenger valves, an air-starting valve, a spray valve and a relief valve. The spray valve atomizes and injects the fuel oil into the cylinder by means of high-pressure air supplied by the engine air compressors. All valves are actuated from a cam shaft located on top of the cylinders, which is operated from the main shaft through a vertical shaft and spiral gears. A pneumatic cylinder automatically places the cam shaft in the proper relation to the valves when the engine is started or reversed. Salt water is used to cool the cylinders of the engine, while the pistons are cooled by fresh water. The water for the pistons is carried through pipes with swinging joints. A small lubricating-oil pump furnishes the oil for the forced lubrication. The main-engine auxiliaries are duplicated, and are independently driven by electric motors.

The two main-engine air compressors are of the two-stage type, and supply high-pressure air for the fuel injection and for replenishing the air in the starting-air flasks. The ship's high-pressure system is connected for use in case of an emergency. The pump sets are independent units, each consisting of a centrifugal salt-water pump, a rotary fresh-water pump, a rotary lubricating-oil pump and a rotary fuel-oil pump. Each set is adequate for full power, the set not in use being a "stand by" set.

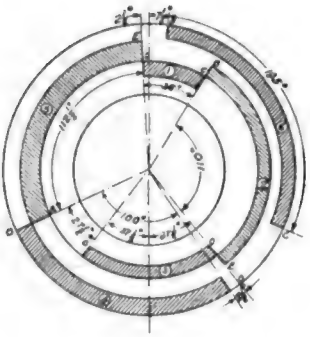


PLATE II.

Plate II shows the valve diagram of the engines.

Main Engine Data.

Type	Nurnberg, reversible, 2-cycle, low-speed, Diesel.
Diameter working cylinders, inches	14.25
scavenging cylinders, inches	22.50
Stroke, inches	23.50
Diameter connecting rods, inches	5.25
Length connecting rods between centers, inches	47.00
Ratio of crank to connecting rod	1 to 4
Diameter of wrist pin, inches	7.00
Length of wrist-pin bearings, inches	7.375
Diameter of main journal, inches	9.00
Unit bearing pressure, with compression at 450 pounds per square inch and combustion pressure at 580 pounds per square inch, wrist pin, pounds	1,409.
crank pin, pounds	1,032.
Volume of working cylinder, cubic inches	3,750.
scavenger cylinder, cubic inches	9,350.
Clearance in working cylinder, cubic inches	314.
scavenge cylinder, cubic inches	1,290.
volume per cent., working cylinder	8.4
scavenge cylinder,	13.8
Linear clearance, working cylinder, inches	2.625
scavenge cylinder, inches	.1885
Cylinder constants, 1, 2, 3, 4, 5 and 6	0.009464
System of cooling cylinders	Salt water.
air compressors	Salt water.
pistons	Fresh water.

Lubrication.

Working cylinders	Forced feed, Detroit oilers.
Scavenger cylinders	Forced feed, Detroit oilers.
Air compressors, both stages	Drip feed.
Bearings, main and crank	Forced lubrication, pump set.

Air Compressor Data.

Type... Two-stage, stepped piston.
Number on engine .... 2.
Diameter first-stage piston, inches.. 11.875
second-stage piston, inches.. 3.875
Stroke, inches 17.75
Cranks are spaced 180 degrees apart, coincident with No. 5 and No. 2 engine cranks, respectively.
Diameter of crank pin, inches 6.75
Length of crank-pin bearing, inches.. 7.0625
Diameter of connecting rod, inches... 3.875
Length of connecting rod, inches .... 53.
Ratio of crank to connecting rod... 1 to 5.97
Unit bearing pressure, pounds, at 1,000 pounds compression, wrist pin.. 548.
Unit bearing pressure, pounds, at 1,000 pounds compression, crank pin.. 247.

SHAFTING.

The shafting is in four sections, consisting of a line shaft in two sections and a crank shaft in two sections. The after section of the line shafting is supported by two bearings, the forward section by one bearing. The bearings are lined with white metal.

Shafting and Bearing Data.

Shaft.	Material.	Length, ft. and ins.	Diameter, inches.	Diam. holes, inches.
Crankshaft, 2 sections....................	Carb. steel	22–05.5625	9.0	3.25
Crank pins, main, engine....................	Carb. steel	0–10.25	9.0	3.25
Crank pins, air compressor..........	Carb. steel	0–07.25	6.75	2.25
Thrust shaft............	Steel	12–10.125	8.0	Solid
Line shaft, 2 sections..................	Steel	33–00	8.0	Solid
Propeller shafting.....	Steel	16–10.5	8.0	Solid

Number of crankshaft bearings.. 10.
Length of main bearings, inches:
Air compressor, No. 1 and No. 2... 7.5625
Main engine, Nos. 3, 4, 5, 6, 7, 8 and 9.................................. 10.
Main engine, No. 10 .... 7.3125

Length of thrust-shaft bearings, inches	8.
Collars on thrust shaft:	
Number	6.
Thickness, inches	1.625
Space between, inches	3.5
Outside diameter, inches	14.
Inside diameter, inches	8.
Bearing surface, square inches	622.
Thrust shoes, number	6.
effective thrust surface, square inches	486.
Line-shaft bearings, number 3, length, inches	10.
Stern-tube bearings—Material, wood:	
Length, forward, inches	17.
after, inches	32.

Propeller Data.

Type	True screw, solid.
Material	Manganese bronze.
Number of blades	3
Diameter, feet and inches	7–09.5
Pitch, feet and inches	6–00
Ratio of diameter to pitch	1.29 to 1.
Area, projected, square feet	14.25
helicoidal, square feet	15.90
disc, square feet	47.68
Height of lower blade above keel, inches	10.
Immersion of upper tip of blade, inches	55.

Condenser and Feed-Heater Data.

Main condenser, Worthington, one located in fireroom; 300 square feet cooling surface.

Feed heater, integral with Roberts boiler.

Evaporators, 36-inch Reilly multicoil, located in fireroom.

Distillers, 18-inch Reilly multicoil, one, No. 7, type D, located in fireroom.

Evaporator feed heater, Wainwright, horizontal, No. 1, type PP and SS.

Fuel-oil heater, one, size OA, single-tube, Schutte and Koerting, spirally corrugated film.

BOILERS.

There are two boilers, one a "Roberts," the other an "Almy" boiler. The original installation contemplated but

one boiler, but experience in service showed the necessity for two. The "Almy" boiler has been installed. The single smoke pipe will care for both boilers. The boilers burn oil with natural draft, are located alongside of each other, and are supplied by the same auxiliaries. Both boilers are water-tube boilers. The top of the smoke pipe is 49 feet 6 inches above the burners.

GENERATORS AND AIR COMPRESSORS.

The generators for charging the submarine batteries are two in number and are driven from the forward end of the main-engine crankshaft. A turbo generator set is used for the ship's lighting and power. There are also two air compressors for charging the air banks of the submarines, and the vessel's air banks, as well as the torpedoes, both electrically driven. Air for pneumatic tools is supplied by a low-pressure compressor.

STEAM AUXILIARIES.

Main air and circulating pump, one; combined, horizontal, simplex, "Worthington;" fireroom.

Main feed pump, one; vertical, simplex, "Blake;" fireroom.

Auxiliary feed pump, one; vertical, simplex, "Blake;" fireroom.

Main sanitary pumps, two; horizontal, duplex, "Blake;" fireroom.

Fuel-oil filling pumps, two; horizontal, duplex, "Blake;" engine room.

Fuel-oil feed pumps, two; horizontal, duplex, "Blake;" fireroom.

Lubricating-oil filling pumps, two; horizontal, duplex, "Blake;" engine room.

Low-pressure air-compressor pump, one; "Westinghouse," vertical; engine room.

Evaporating and distilling-plant pumps:

Distiller fresh-water, horizontal duplex; fireroom.

Evaporator feed, horizontal, duplex; fireroom.

Distiller circulating, horizontal, duplex; fireroom.

Steering engine, one; automatic worm-gear type, American Engineering Company, Phila., Pa.; second deck.

Anchor engine, one; Providence spur-gear type, American Engineering Company, Phila., Pa.; main deck, forward.

Towing engine, one; Automatic Steam Towing Machine, American Engineering Company, Phila., Pa.; main deck, aft.

ELECTRICAL AUXILIARIES.

Main air compressors, two; 90 H.P., open, General Electric Co.; engine room.

Main circulating pumps, two; 7 H.P., enclosed, Electro Dynamic Co.; engine room.

Main generators, two; 300 k.w., open, Crocker and Wheeler; engine room.

Forge blowers, one; 0.125 H.P., enclosed, Emerson Electrical Manufacturing Co.; workshop.

Machine-shop motor, one; 20 H.P., enclosed, General Electric Co.; workshop.

Fire and bilge pumps, one; 25 H.P., enclosed, and ventilated, General Electric Co.; engine room.

Ice machine, one; 5 H.P., open, General Electric Co.; fire-room.

Furnace and oil pump, one; 1.25 H.P., open, Westinghouse Electric and Manufacturing Co.; workshop.

Deck winch, one; 20 H.P., enclosed, General Electric Co.; upper deck.

Ventilating blowers, two; 1.125 H.P., semi-enclosed, B. F. Sturtevant; second deck.

Auxiliary generator, one; 35 k.w., open, turbo, General Electric Co.; engine room.

SHOP EQUIPMENT.

Machine Shop.—

1 screw-cutting back-geared extension-gap lathe, 7 feet between centers when extended, swing 32 inches.
1 tool-room lathe, 14.5-inch swing, 21 inches between centers.
1 screw-cutting back-geared tool-room lathe, 14.5-inch swing, 57 inches between centers.
1 8-inch back-geared precision lathe, 22 inches between centers, with milling attachment and grinders.
1 16-inch high-speed sensitive drill fitted for No. 2 Morse taper.
1 upright drill, up to 1.5 inches to 14 inches from edge of work, 9-inch traverse of spindle, fitted for No. 4 Morse taper.
1 column tool-room shaper, 15-inch stroke, 20-inch traverse.
1 double emery grinder on column, wheels 12-inch diameter, 2-inch face.
1 universal milling machine, with overhanging arm, 20-inch longitudinal feed, 17-inch vertical movement, 7-inch traverse, spindle fitted for No. 10 Browne and Sharp taper.
1 power hack saw, to take up 8-inch solids.

Woodworking Tools.—

1 pattern-maker's gap lathe, 36-inch swing through gap, 4 feet between centers when extended.
1 band saw, 32-inch.
1 hand-operated universal wood trimmer, on column.

Foundry and Blacksmith Shop.—

1 marine combination furnace, oil-burning, to melt brass in a crucible, to serve as forge for blacksmith and coppersmith.
1 blacksmith forge, 42 inches square, 24 inches high, with electric-driven air blast and tuyere blast gate.

TRIALS.

The contract required the following trials:

(*a*) A progressive trial over a measured-mile course at Provincetown, Mass., for standardizing the screw, extending from maximum speed, at least 12.25 knots, down to a speed of 8 knots.

(*b*) A full-speed trial of eight hours duration in the open sea, in deep water, at the highest speed attainable, the average for the eight hours to be not less than 12.25 knots.

(*c*) A fuel-consumption trial of four hours in the open sea, in deep water, at an average uniform speed of 11 knots, as nearly as possible, all auxiliaries in operation, the consumption to be such that the vessel will have a cruising radius of 3,400

TABLE I—U.S.S. FULTON
STANDARDIZATION TRIAL DATA
OCTOBER 31, 1914.

No. of Run	Time on Course: Mins.	Time on Course: Secs.	Speed in Knots	R.P.M.	I.H.P.
5	8	49	6.81	141.3	766
6	7	43.3	7.77	146.6	840
7	8	9.6	7.35	148.3	688
Mean of Group			7.425	145.7	784
8	6	4.8	9.87	188.9	556
9	6	19.8	8.48	186.0	573
10	6	13.4	9.64	185.9	560
Mean of Group			9.618	186.7	566
11	5	26.4	11.03	221.5	694
12	5	21	11.21	224.4	758
13	5	17.9	11.32	224.5	—
Mean of Group			11.193	223.7	726
14	5	1.7	11.93	240.7	982
15	4	58.8	12.05	241.9	1012
16	5	2.4	11.90	240.7	912
Mean of Group			11.983	241.3	980
17	4	43	12.72	258.0	—
18	4	42	12.77	260.8	1098
19	4	40.8	12.82	259.2	1042
20	4	39.7	12.87	260.8	1044
21	4	44.9	12.64	257.6	994
Mean of Group			12.785	259.7	1051

Data on Steaming Trials—Table II.

Average results.	3.5-hour 11 knots.	8-hour, full speed.
Speed, knots	11.092	12.342
R.P.M., number	221.41	249.5
I.H.P.	831.	1,097.
Fuel-oil consumption per hour, main engine only, pounds	350.5	483.3
Fuel-oil consumption per hour, all machinery, lbs.	622.5	652.2
Fuel-oil consumption per knot at speed indicated, pounds	56.12	39.16
Fuel-oil consumption per knot guaranteed, pounds	73.5	41.5
Radius of action on fuel oil at indicated speed, knots.	5,372.5	5,706.
Lubricating oil consumed, gallons	10.	13.
Radius of action on lubricating oil at indicated speed, knots	10,404.	20,351.
Turbo generators—Ampères	120.	53.4
Volts	120.	120.
R.p.m.	3,500.	3,587.5
Pressures—Boilers, pounds	154.	149.5
Feed, pounds	164.4	154.1
Oil to burners, pounds	136.4	144.3
Sanitary system, pounds	24.3	26.9
Vacuum in condenser, inches	16.4	17.4
Temperatures, degrees F.—Engine room	61.1	67.4
Fireroom	71.1	74.8
Outside air	55.	62.1
Feed tank	not taken.	123.3
Barometer, inches mercury	30.04	30.00
Double strokes or r.p.m.—After feed pump	18.	10.4
Fuel-oil pump	8.	8.5
Sanitary pump	50.3	50.6
Air and circ. pumps	28.3	22.00
Distiller circ. pump	60.6	......
Ice machine	120.	......
Pressures, lbs. per sq. in.—Evaporator steam	25.	......
shell	3.	......
Spray bottle	1,000.	1,076.3
Starting bottle	828.5	845.6
1st stage air comp., for'd.	76.4	43.1
aft	54.3	56.1
Scavenger receiver	8.	9.1
Low-pressure fuel pumps.	1.4	1.9
Circulating water	8.9	11.3
Lubricating oil	24.9	24.7
Fresh water	10.	12.1
Temperatures, deg. F.—Scavenger receiver	122.	136.5
Circ. water to engine	58.	56.6
from engine	89.	94.3
Fresh water to engine	76.7	76.1
cooler	104.	113.4
Circ. water from FW., cool.	88.	91.8
Lubricating oil to engine	71.	74.9
cooler	82.	87.6
Circ. water from oil cooler.	60.	59.9
R.p.m. of independent pump aft	1,507.	1,466.3
Main generators.—Volts	120.	121.1
Ampères	57.	50.
Water evaporated during trial, gallons	375.	......

nautical miles, exclusive of the fuel oil carried for the submarines.

(*d*) Trials to determine the ability of the vessel to back satisfactorily, to test the steering gear, and to test the anchor gear. Reversing tests, the time to reverse not to exceed 15 seconds.

Trials were held October 31, and November 1, 1914, the data for which is given in Tables I and II. From the data in Table I the standardization curve, Plate III, was plotted.

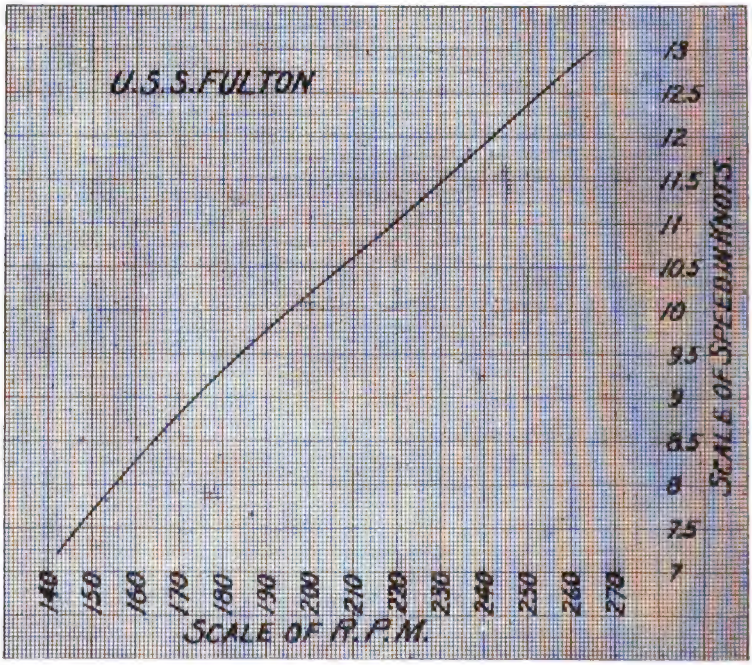


PLATE III.

A DEVELOPMENT OF A HIGH-GRADE ALLOY STEEL AT LOW COST.

BY LIEUTENANT J. B. RHODES, U. S. NAVY, MEMBER.

1. It is the purpose of this paper to describe the qualities of an alloy steel which has been found to furnish high-grade castings and forgings from the same mix and at comparatively low cost. The principles involved are comparatively well known and no originality is claimed, but it is believed that the alloy described (and its complement) is being regularly manufactured for use for the first time.

2. In order to increase the output of a small open-hearth furnace it was desired to manufacture ingots for forging purposes without abnormally increasing the cost of castings. These ingots were to be used for forgings which are required to show minimum physical values in a transverse direction, as follows:

Tensile strength, 95,000 pounds per square inch.
Proportional limit, 65,000 pounds per square inch.
Elongation, 18 per cent.
Reduction of area, 30 per cent.

It was believed that the steel as cast would, if capable of showing the above values in forgings, give in properly annealed castings the following physical values:

Tensile strength, 85,000 pounds per square inch.
Yield point, 53,000 pounds per square inch.
Elongation, 22 per cent.
Reduction of area, 35 per cent.
Bend, 120 degrees around 1-inch diameter.

3. In going over the ground of previous experimenters the following points were determined:

(*a*) In order to obtain a steel of high physical properties with considerable ductility and machinability it is necessary to produce a pearlitic steel. High-grade steels are amorphous to a great degree and are fine grained. The microscope should show a uniform mixture of constituents. Although it is generally considered that true pearlitic is a eutectic of cementite and ferrite, it is known that a pearlitic structure is obtained in alloy steels in which the amount of pearlite is in excess of that due to carbon alone.

(*b*) Knowing that brittle ranges are found where carbon, nickel, chromium, manganese, etc., are present in the percentages necessary to give true eutectics, and knowing the superiority of ternary and quarternary steels, *i.e.*, steels containing chromium and nickel and those containing chromium, nickel and vanadium in addition to the carbon, it was decided to limit the percentage of each hardening or toughening element and to increase the pearlite by the use of additional elements. The following were considered:

(*a*) Carbon, which increases both strength and hardness, but which should not be present in larger amount than 0.60 per cent.

(*b*) Manganese, which increases the strength and hardness and acts as a deoxidizer, but which should not be present in larger amounts than 2 per cent.

(*c*) Nickel, which increases strength and toughness, but which should not be present in larger amounts than 5 per cent.

(*d*) Copper, which has an effect similar to that of nickel, but should not exceed 4 per cent.

(*e*) Chromium, which hardens the metal, increases its susceptibility to heat treatment, but which should not exceed 3 per cent.

4. Consideration of the above shows the possibility of obtaining steel of the grade desired by adding copper and manganese to the usual nickel-steel alloy. Steel containing more of the hardening agents than shown above are liable to

be brittle and treacherous; in fact, the amounts given above are considered rather high. Of these alloying agents carbon and manganese are the most easily obtained and the least costly. It was decided to limit the carbon to 0.35 per cent. and the manganese to 1.20 per cent., or about 0.60 per cent. higher than normal. It was found that nickel could be obtained in turnings from 3 per cent. nickel-steel in sufficient quantity to give 1 per cent. to 1.5 per cent. in the steel, and nickel with copper could be obtained in the form of monel-metal scrap and turnings containing approximately 65 per cent. nickel and 30 per cent. copper.

5. The increase in manganese by 0.6 per cent. or 6 pounds per 1,000 can be made by adding not more than 10 pounds per 1,000 of manganese, which in the form of 80 per cent. ferro-manganese, costs about $0.06 per pound for manganese, an increase of 60 cents per 1,000 pounds, or $0.006 per pound. Nickel-steel turnings are worth about $10.00 a ton more than ordinary scrap (as based on sale of own scrap), which means that nickel in this form is obtained for $0.05 per pound instead of $0.35 or $0.40 per pound. To obtain 1.0 per cent. to 1.5 per cent. nickel 15 pounds per ton are added, at a cost of 75 cents per 1,000 pounds, or $0.00075 per pound. To obtain .50–.75 copper, monel-metal scrap at $0.12 per pound is added. Two per cent. of monel metal gives .70 copper and 1.20 nickel, at a cost of $2.40 per 1,000 pounds, or $0.0024 per pound. The cost of these additions are as follows:

Manganese	$0.0006
Nickel	0.00075
Monel	0.0024
Total	$0.00375

So that for an increase of less than $0.004 per pound we obtain a steel equal in properties to a 3 per cent. nickel-steel. The composition of the steel may be taken to be as follows:

Carbon, . .	.30 per cent. to .35 per cent.,
Silicon, . .	.25 per cent. to .35 per cent.,
Phoshorus and sulphur, .	Not over .05 per cent.,
Manganese, . .	1.00 to 1.20 per cent.,
Nickel, . . .	1.50 to 1.80 per cent.,
Copper, . .	.50 per cent. to .80 per cent.,

which steel will show properties equal to those specified.

6. It is believed that the excess manganese prevents red shortness due to copper oxide by combining with any oxygen present in the bath. Castings are remarkably free from checks, cracks, blowholes and shrinks, and ingots are normal. The addition of chromium in sufficient quantity to give 0.50 per cent. chromium in the alloy increases the physical properties to those of ordinary chrome-nickel steel containing 1 per cent. chromium and 3 per cent. nickel.

7. The following table shows some physical tests and analyses of castings and forgings:

TEST BARS FROM COUPONS CAST ON BODY OF CASTINGS.

Heat No.	C.	S.	Mn.	Si	P.	Ni.	Cu.	Tensile.	Y. point.	Per cent. elong.	Per cent. red.	Bend deg.
201	.36	.036	.81	.31	.034	1.60	.76	92,283	55,767	22	37	120
204	.30	.037	.93	.36	.04	1.44	.85	92,334	55,003	21	31	120
207	.33	.030	1.02	.38	.035	1.15	.49	88,616	54,494	25	40	120
208	.34	.034	1.04	.36	.033	1.14	.52	92,691	56,277	22	34	120
214	.35	.035	1.11	.33	.042	1.34	.60	95,543	59,078	22	37	120
215	.30	.032	.93	.35	.043	1.12	.77	90,042	57,550	23	37	120
216	.27	.04	1.18	.35	.046	1.60	.61	92,691	55,003	22	36	120
218	.34	.042	1.15	.35	.035	1.58	.69	94,830	54,748	22	36	120
219	.27	.035	1.02	.30	.047	1.47	.42	85,510	56,022	23	34	120
223	.38	.029	1.02	.35	.049	1.10	.80	88,718	55,512	23	38	120
228	.30	.031	1.15	.38	.046	1.60	.56	96,510	59,587	23	37	120
230	.36	.036	1.18	.31	.049	1.37	.80	94,117	50,929	23	36	120
232	.36	.036	1.03	.35	.039	1.47	.63	90,144	55,225	23	34	120
233	.36	.035	1.03	.40	.04	1.73	.21	96,052	59,087	25	41	120
234	.32	.036	1.21	.36	.049	1.24	.65	91,366	55,003	25	44	120
236	.31	.032	1.15	.34	.043	1.54	.70	81,343	60,096	25	41	120
241	.31	.033	.96	.39	.042	1.27	.64	87,343	53,457	23	36	120
245	.31	.032	1.15	.37	.037	1.80	.62	92,181	55,288	23	36	120
248	.33	.032	.79	.42	.048	1.69	.65	87,191	54,239	23	38	120

Test Bars Cut From Body of Forgings—Transverse.

(*Steel of same heats shown above.*)

Heat No.	Tensile strength.	Limit of proportionality.	Per cent. elongation.	Per cent. reduction.
A-410-6	102,469 98,853	72,421 64,272	21.65 23.80	37.6 47.3
A-410-9	99,719 97,529	63,254 63,254	21.85 26.45	42.2 52.4
A-410-10	100,585 97.529	68,347 63,254	17.35 26.45	30.78 52.4
A-408-1	110,516	77,412	21.6	45.2
A-408-2	113,164	80,570	18.9	34.08
A-408-3	110,923	77,514	23.5	51.3
4 × 1	117,952	85,561	22.3	51.0
4 × 2	119,836	89,890	20.2	49.6

Test Bars Cut From Body of Forgings—Longitudinal.

A-779-1	105,423	72,319	24.8	56.4
A-779-4	95,135	67,226	28.7	65.7

Forgings of Same Composition Carrying About 0.50 Per Cent. Chromium.

A-749	130,379	117,608	19.0	47.6 Long.
6 × 1-1	141,940 146,417	107,050 113,739	16.0 15.2	42.0 Trans. 45.0 Trans.
6 × 1-2	156,329 147,695	117,429 108,021	15.5 15.1	42.5 Trans. 42.2 Trans.
6 × 3 6 × 4 6 × 5	126,151 130,328 130,226	106,543 106,441 110,516	16.6 17.7 19.6	40.0 Trans. 43.0 Trans. 48.5 Trans.

DESCRIPTION AND TRIALS OF U. S. S. *SACRAMENTO*.

(*Gunboat No. 19*).

BY W. F. SICARD, ASSOCIATE.

The construction of the *Sacramento,* Gunboat No. 19, was authorized by an Act of Congress, approved March 4, 1911.

Plans and specifications for this vessel were issued to prospective bidders, May 1, 1912, and bids were opened at the Navy Department, at noon, July 1, 1912.

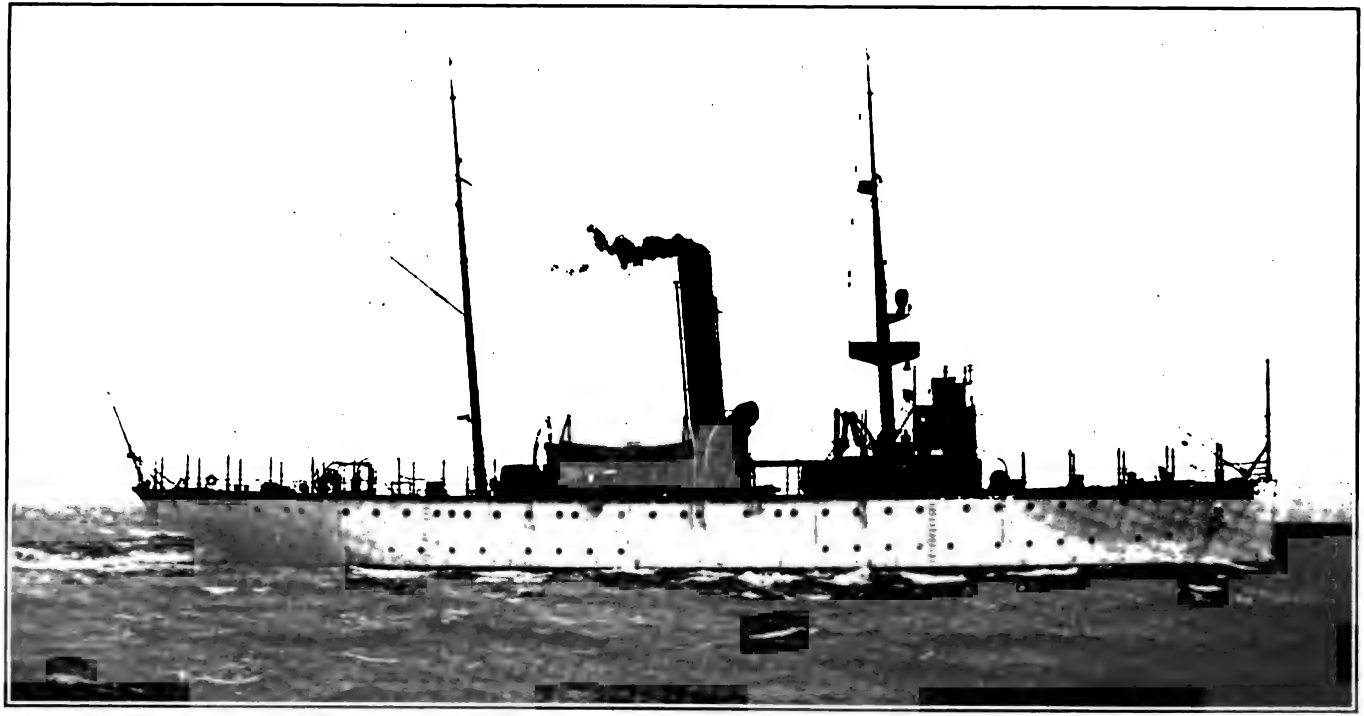
The contract, awarded to The William Cramp and Sons S. & E. B. Co., of Philadelphia, Pa., at a price of $492,500.00, called for a vessel of about 1,425 tons trial displacement. It contained the usual provision embodied in the act entitled "An act to increase the naval establishment, of August 3, 1886." The vessel was to be completed and delivered to the Government within 21 months of the date of contract and there were the usual penalties for under speed and over time.

The following trials were required by the contract:

(*a*) A progressive trial over a measured course at Delaware breakwater, for standardizing the screws, extending from maximum speed down to a speed of 8 knots, about 14 runs over the course being specified. On this trial the guaranteed speed for the five highest runs was 12½ knots.

(*b*) A full-speed four-hour trial, in deep water in the open sea, at the highest speed attainable, the speed developed upon this trial to be not less than an average of 12½ knots an hour.

(*c*) A coal-consumption trial of 10 hours in the open sea in deep water, at an average uniform speed of 10 knots.



U. S. S. "SACRAMENTO" (GUNBOAT NO. 19).

(*d*) Trials to determine the ability of the vessel to back satisfactorily at full speed and at cruising speed; trials for stopping, and the usual trials for the steering gear and the anchor engine.

The guaranteed coal consumption per knot run, including coal necessary for all auxiliaries in use on trial, was not to exceed 155 pounds on trial (*b*) and 127 pounds on trial (*c*).

HULL DATA.

The *Sacramento* is a single-screw gunboat, built of steel. She has flush main and berth decks, and has forward and after orlop decks. The principal dimensions and characteristics of the hull, are as follows:

Length between perpendiculars, feet	210–00
on load water line, feet	210–00
over all, feet and inches	226–02
Beam, extreme, feet and inches	40–10⅜
at load water line, feet and inches	40–10⅜
Ratio length to beam	5.13
Draught, forward, feet and inches	11–06
aft, feet and inches	11–06
mean, feet and inches	11–06
Displacement, load draught, tons	1,431.47
per inch at L.W.L	12.94
Area, immersed midship section, square feet	437.0
L.W. plane, square feet	5,437.5
Wetted surface, square feet	9,180.
Coefficient of fineness, block	.507
midship section	.950
L.W.L.	.630

Masts.—There are two steel masts, each about 105 feet 5¾ inches above the water line, and located at frames Nos. 24½ and 51½. They are fitted with wireless, signal yards, etc. The foremast has a searchlight platform 51 feet 6¾ inches above water line and there is a military top, 43 feet 6¾ inches above waterline. Both masts are fitted with grab-irons, for climbing.

Navigating Bridge.—The navigating bridge is immediately

above the chart house and extends from frame 19 to frame 22. There is a ladder on the after end, starboard side, leading to the superstructure deck.

Superstructure Deck and Chart House.—This deck extends between frames Nos. 19 and 28 and contains the chart house, from frames Nos. 19 to 23. On each side of the superstructure deck is mounted a 3-pdr. saluting-gun. The foremast passes through this deck a little forward of frame No. 26.

Main Deck.—The main deck, on which is located the battery, is a weather deck throughout. There are two deck-houses, one forward, between frames Nos. 19 and 28, containing the conning tower, and captain's cabin, stateroom, bath and pantry, the other between frames Nos. 34 and 42, containing smokepipe and ventilator inclosures, bread room to port and radio room to starboard, while the officers' and crew's galley occupy the after part.

Berth Deck.—The berth deck extends from stem to stern. Beginning forward, there is the paint room, shower and bathroom for wardroom officers and on each side seven wardroom officers' staterooms, the wardroom country being located at center of ship, between the staterooms mentioned. Next to the above, abaft frame No. 19, is the wardroom mess room. extending to frame No. 23, this is followed by the wardroom pantry, outboard of which, on the starboard side, is the navigator's stateroom, the executive officers' stateroom and the captain's and executive officers' office, while in similar locations, on the port side, is the paymaster's stateroom, the senior engineer officer's stateroom and the pay office. The above enclosures end at frame No. 31, and from here to frame No. 45 is crew's space, broken only by the boiler hatch, at center of ship, and in wake of this, on the port side, the firemen's washroom, with ladder leading down to the fireroom. The engine-room hatch extends between frames Nos. 42 and 48, and, on the starboard side, between frames Nos. 47 and 64, is the engineer's office, chief petty officers' washroom, and ship's stores, and, on the port side, the operating room, sick bay

and sick-bay bath. The above subdivisions end at bulkhead No. 54, and forward of this bulkhead, at center of ship, is the armory. The remainder of berth deck, abaft frame No. 54, is crew's space, crew's water closets and urinals and the prison.

Orlop Deck.—The orlop deck extends from stem to stern, being broken only to accommodate the coal bunkers, the boiler room and the upper engine room. On the starboard side, outboard of the engine room is the engineer's workshop and, similarly located, on the port side, is space for 12 chief petty officers. The remainder of this deck, aft, is crew's space, except for the refrigerating room, extending from frame No. 60 to 63.

Hold.—In the hold, from forward, aft, is the forward trimming tank, chain locker, provisions, magazine, stores, coal bunkers, boiler compartment, engine compartment, shaft alley, ammunition rooms, stores and after trimming tanks. The boiler compartment extends between frames Nos. 34 and 41, and the engine compartment between frames Nos. 41 and 48.

Inner Bottom.—The reserve feed-water tanks are in the inner bottom, the full width of the boiler compartment, and extend between frames Nos. 34 and 37.

COMPLEMENT.

The ship's complement will be approximately as follows:

Commanding officer	1
Wardroom officers	9
Chief petty officers	12
Seamen's branch	73
Artificer branch	10
Artificer branch (engine-room force)	33
Special branch	7
Commissary branch	4
Messmen branch	11
Total	160

BATTERY.

The battery consists of the following:

One 4-inch rapid-fire gun	Main deck.
One 4-inch rapid-fire gun	Main deck.
One 4-inch rapid-fire gun	Main deck.
Two 3-pdr. saluting guns	Superstructure deck.
Two 1-pdr. guns	Main deck.

SMALL BOATS.

The following boats are carried, all on main deck except the whale boats, which are carried in davits:

Two 30-foot steamers.
Two 30-foot motor sailing launches.
Two 28-foot whale boats.
One 16-foot dinghy.
One 12-foot wherry.

Coaling Gear.—The coaling gear consists of one electric winch with two gypsy heads, capable of lifting 1,600 pounds at a speed of 125 feet per minute, and four coaling booms of 1,600 pounds capacity each.

The capacity of the coal bunkers is as follows:

B–2, Port side wing bunker	53 tons.
B–3, Starboard side wing bunker	56 tons.
A–18, Port, athwartship bunker	91 tons.
A–19, Starboard, athwartship bunker	91 tons.
A–16, Port, forward bunker	65 tons.
A–17, Starboard, forward bunker	65 tons.
Total	421 tons.

Fire Main.—The fire main is supplied by the fire and bilge pumps in the engine room and the fireroom. It extends, on the orlop deck, through the machinery spaces forward to frame No. 19 and aft to frame No. 59. There are 3½-inch

risers at frames Nos. 19, 35½, 42½ and 60, and, in the engine room, an emergency connection to supply circulating water to the distillers. At frame No. 19 the fire main rises to the berth deck with a 2½-inch fire plug at frame No. 19, a 3½-inch riser at frame No. 16, a 2-inch fire-main relief valve forward of this riser, and a connection to deck scupper pipe at frame No. 12½. The riser at frame No. 35½ extends to the berth deck with 2½-inch plugs at frame 31½ and a 2½-inch riser, to deck above, at frame 28.

HULL AUXILIARIES.

Steering Engine.—There is a steering engine, located in the after end of the engine room. It is a double engine with two 9-inch diameter cylinders and 6-inch stroke. It is manufactured by the American Engine Co., of Philadelphia, Pa.

Anchor Engine.—The anchor engine, located forward on the orlop deck, is also manufactured by the American Engine Co. It is a double engine with 6-inch diameter cylinders and 8-inch stroke.

Sanitary Pumps.—There are two sanitary pumps, located on the port side of the engine room, for the sanitary and flushing system. They are of the centrifugal, electric motor-driven type, and each is capable of operating the system. Each has a capacity of 250 gallons of water at 1,500 r.p.m. and is driven by a 7½-H.P. General Electric Co. motor. The suction is 4 inches and the discharge 3 inches in diameter.

Fresh-Water Pumps.—There are two fresh-water pumps, for ship's purposes. They are of the centrifugal type and electric driven. Each has a capacity of 70 gallons of water at 1,800 r.p.m. and is driven by a 3-H.P. General Electric Co. motor.

Main Drain.—The main drain, 7 inches diameter, runs from the boiler compartment in a single pipe, on the port side, to the after end of the engine room, where it is connected to the main circulating-pump suction pipe. There are 7-inch diam-

eter valves at the bilge wells in engine and boiler compartments, operated in place and from the berth deck. There is also a valve at the connection with the circulating-pump suction pipe, as a safeguard against flooding the main drain from the sea.

Secondary Drain.—The secondary drain extends in a single line from a manifold at frame No. 25 to the after trimming tank. It is 5 inches diameter to the after end of the engine room, abaft which it is 4 inches in diameter to the trimming tank. The manifold at frame No. 25 has a 3-inch diameter suction from forward trimming tank, four 3-inch suctions from holds and a 5-inch suction from coal bunker. In the fireroom there is a 5-inch bilge suction, two 5-inch suctions from coal bunkers and a 4-inch connection to the fireroom fire and bilge pump. In the engine room there are two 3-inch connections to the double bottom, a 5-inch bilge connection, and a 4-inch connection to the engine-room fire and bilge pump. From the after end of the engine room the system consists of a single pipe, 4 inches diameter, leading through the shaft alley to the after trimming tank, with 3-inch branches to the hold, after trimming tank and shaft alley. The shaft-alley connection is fitted with a Macomb strainer. At the after end of the engine room there is a 5-inch suction, through a Macomb strainer, from the bilge.

Heating System.—The ship's heating system is arranged in five circuits, as follows:

No. 1, Officers' quarters and heaters of ventilating system No. 4.

No. 2, Heaters of ventilating systems No. 2 and No. 5.

No. 3, Crew's quarters (radiators).

No. 4, Galley, pantry and plumbing heaters and fresh-water gravity tanks.

No. 5, Sterilizing apparatus, and plumbing heaters in operating room.

All pipes are of drawn brass, iron-pipe size. Coils are of 1-inch diameter pipe.

MAIN ENGINE.

There is one main engine, designed to develop 950 indicated horsepower when making 125 revolutions per minute. The engine is of the direct-acting, vertical, inverted, three-cylinder, triple-expansion type, the engine turning to starboard when going ahead. The order of cylinders, beginning forward, is high-pressure, intermediate-pressure and low-pressure. The cranks are at angles of 120 degrees.

Cylinders.—The cylinders, valve chests and cylinder and valve-chest covers are of cast iron. The cylinders are not steam jacketed. The high-pressure and intermediate-pressure valve-stem guides are cast with the lower valve-chest covers, and the low-pressure valve-stem guide is bolted to the under side of its valve chest. The intermediate-pressure and low-pressure balance cylinders are cast with the upper valve-chest covers. The valve-stem guides are fitted with composition bushings.

Framing.—Each cylinder is supported by a cast-iron, inverted Y-shaped box frame and a Class B forged-steel column, 4½ inches in diameter. Each box frame is secured to the bedplate and to its cylinder by 1⅛-inch diameter bolts, 16 and 6 respectively. The columns are secured at bottom and top by four bolts, 1⅝-inch diameter and 1½-inch diameter respectively. The steel columns are on the working side of the engine. The Y frames carry facings for securing the crosshead guides and the reversing-shaft brackets.

Crosshead Guides.—The crosshead guides are of the bar type, of cast iron and hollow, for the circulation of cooling water. At the upper end each is bolted to a facing on its cylinder by two 1¼-inch-diameter fitted bolts, and at the lower end to a facing on the box column by two 1¼-inch fitted bolts.

Bedplate.—The bedplate is of the box type, of cast iron, in one casting, and supported on keelson plates. Proper facings are provided for main bearings, columns, box frames, etc.

Main Bearings.—There are six crank-shaft bearings, one

pair to each crank, and each has two 2¼-inch diameter cap bolts. Each bearing is composed of a bottom brass, two distance pieces, and a cap. The cap is of cast steel, Class B, and the other parts of composition. The cap and brass are lined with white metal and provided with oil channels. The bottom brass is cored for the circulation of cooling water.

Pistons.—All pistons are of the conical type, of cast-steel Class B and are fitted with cast-iron followers, bull rings and packing rings. The pistons have two packing rings cut obliquely in one place, and clamped by fitting into a recess in the bull ring. No piston rings are fitted.

Piston Rods.—The piston rods are Class B forgings, solid, and are accurately ground and polished. The upper end is tapered to fit the piston which is secured by a collar, nut and split key. A dowel pin, between piston and rod, prevents turning. The lower end is also tapered and secured to the crosshead by a shoulder, nut and a split pin.

Crossheads.—The crossheads are Class B forgings, secured to taper on lower ends of piston rods. Each is secured to the flange provided on the crosshead slipper by four 1⅛-inch-diameter stud bolts. The crosshead pins have a 1-inch-diameter axial hole extending from outer face to body of crosshead, and the pins are flattened 3/16 of an inch, on each side, to assist in retaining the lubricating oil.

Crosshead Slippers.—Crosshead slippers are of manganese-bronze and are faced with white metal. Each slipper grasps three sides of its bar guide and is secured in place, on the fourth side, by means of two liners and a manganese-bronze cap, through the medium of four 1⅛-inch diameter bolts. The effective ahead surface of the slipper is 9 by 15½ inches and the backing surface is 9 by 14 inches.

Connecting Rods.—The connecting rods are solid, of forged-steel, Class B, forked at the top to span the crosshead and "T" headed at the bottom to receive the crank pin box. The crosshead boxes are composition with composition liners and the crank pin boxes are lined with white metal.

Valve Gear.—The valve gear is of the Stephenson type, with double-bar links, and all valves are worked direct. There is one single-ported valve for the high-pressure cylinder, one for the intermediate-pressure cylinder and a double-ported slide valve, working on a false valve seat, for the low-pressure cylinder. The high-pressure valve rings are solid; the intermediate-pressure valves are fitted with bull rings, each bearing two narrow rings, cut obliquely. The intermediate and low-pressure valves are each fitted with the ordinary type of balance piston. The back of the low-pressure valve is provided with packing bars of hard cast iron, working against a relief frame. There is no independent cut-off gear other than the adjustable blocks in the reversing arms.

Eccentrics and Eccentric Straps.—The eccentrics are of cast iron, 3⅜ inches wide, each in two parts and bolted together by two 1⅛-inch-diameter bolts. They are rabbeted on each side for the flanges of the eccentric straps. The eccentrics are keyed to the shaft. The eccentricity is 2½ inches. The eccentric straps are of composition, lined with white metal ⅜ inch thick, dovetailed and hammered into place. The straps are in halves, secured together by two 1⅛-inch-diameter bolts. The eccentric rods, of forged-steel, Class B, have forked ends for spanning the links and "T" heads for securing to the eccentric straps. Each is secured by two 1⅛-inch-diameter studs. The link ends of rods are fitted with caps Class B, forged-steel, liners and composition bushings, and are adjustable. The eccentric rods are 1¾ inches diameter at top and 2½ inches diameter at bottom. The suspension links are 1 inch diameter at ends and 1¼ inches diameter at middle, and are of forged-steel, Class B. They are fitted with Class B forged-steel caps, liners and composition bushings and are adjustable. The link bars and blocks are Class B forgings and the gibs are of composition H.

Reversing Gear.—The reversing gear is of the vertical, direct-acting type, and fitted with follow-up gear. It is bolted

to the high-pressure cylinder of the main engine and is connected, through a connecting rod, to the reversing-shaft arm, the reversing shaft being connected to the main links through the reversing arms and the suspension links. The reversing-engine cylinder is 6 inches diameter by 11 inches stroke. The gear is controlled by a floating lever, the primary motion being taken from a hand lever at the working platform, the secondary motion being taken from the reversing arm.

Reversing Shaft.—The reversing shaft is solid, of forged-steel, Class B, 3¾ inches diameter. It is supported by double cast-iron brackets, secured to the high-pressure and intermediate-pressure columns. The reversing lever and reversing arms are of cast steel, Class B, the reversing arms being slotted for the reception of the blocks and graduated, the blocks being adjustable by means of a handwheel and screw, for altering the cut-off.

Turning Gear.—Provision is made for turning the main engine by hand.

Lubrication.—Lubrication is accomplished through distributing-oil boxes, located on the main engine cylinders. There is a 3½ by 1¾ by 4-inch steam pump, in the engine room, for keeping the distributing oil tank supplied with oil. This pump draws from the storage tanks and discharges to the distributing tank. The distributing oil boxes have connections to distribute the oil to the various bearing surfaces and are plainly marked with the names of the parts to which they lead. Each connection has an adjustable valve, a sight feed with glass protecting tube, and a wick tube. There is an electric-light installation behind the sight feeds.

Water Service.—There is a 1¼-inch-diameter water-service main, attached to the main engine, having a 1-inch-diameter branch to the thrust bearing, a ¾-inch-diameter branch with a hose connection for each crank-shaft bearing, two ¾-inch-diameter pipes for each crank pin, and a ½-inch-diameter branch to each hollow brass in crank-shaft bearings.

Main Engine Data.

Working pressure at H.P. chest, pounds by gage	200
Revolutions per minute, designed	125
Indicated horsepower of main engine, designed	950
Diameter of H.P. cylinder, inches	16
I.P. cylinder, inches	26½
L.P. cylinder, inches	44
Stroke, inches	26
Ratio, I.P. to H.P	2.74
L.P. to H.P	7.56
L.P. to I.P	2.75

Cylinder clearances:	Per cent. of volume. Top.	Per cent. of volume. Bottom.	Linear, inches. Top.	Linear, inches. Bottom.
High-pressure	18.2	16.7	5/16	⅜
Intermediate-pressure	15.7	14.0	⅜	½
Low-pressure	9.2	9.3	25/64	9/16

H.P. valve, piston, diameter, inches	8
I.P. valve, piston, diameter, inches	14½
L.P. valve, double ported slide, length, inches	30¾
width, inches	42
Valve stems, diameter at stuffing boxes, inches	2⅛
below stuffing boxes, inches	1⅞
through valves, inches	1½
Balance pistons, I.P. diameter, inches	3½
L.P. diameter, inches	8
Piston rods, diameter, inches	4
length, surface to surface, feet and inches	3–11¼
Cylinder walls, H.P., I.P., and L.P., thickness, inches	1¼
Valve-chest liners, H.P., thickness, inches	⅞
I.P., thickness, inches	1
Cylinder relief valves, one each end each cylinder, H.P. diameter, inches	1½
I.P. and L.P. diameter, inches	2
Connecting rod, length center to center, feet and inches	4–10½
diameter at upper end, inches	3⅝
at lower end, inches	4⅜
crosshead end bolts (4), diameter, inches	2
crank-pin end bolts (2), diameter, inches	2¾

SHAFTING AND BEARINGS.

There is one line of shafting consisting of crank shaft, thrust shaft, two lengths of line shaft and one propeller shaft. All shafting is of forged-steel, Class B. Coupling bolts are of forged-steel, Class A.

Valve Diagram Data.—Normal.

Name of vessel	*Sacramento.*
Total I.H.P.	950 at 125 r.p.m.—200 pounds steam pressure.
Type and number of engines	One vertical, triple-expansion.
Diameter of cylinders and stroke	16″×26½″×44″; stroke 26″; ratio 1 : 2.74 : 7.56.
Valve gear	Stephenson link, double bar, direct connected, crossed rods.
Connecting rod	4′-10½″ C. to C. 4½ cranks.

	H.P.		I.P.		L.P.	
Eccentricity, inches	2½		2½		2½	
Travel of valve, inches	5		5		5	
Kind and No. of valves. Diameter of piston	one 8″ P.V.		one 14½″ P.V.		D.P.S.V.	
Steam taken on	Inside.		Outside.		Outside.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of port and clear length, inches	1½ × 19.1328		1¾ × 36.55		2′-2″ × 36	
Steam lap, inches	1 7/32	1 9/64	1 5/64	1	1 3/64	61/64
Exhaust lap, inch	0	+ 1/8	0	+ 15/32	0	+ 27/64
Angular advance, degrees	37½		35½		36½	
Steam lead, angular, degrees and minutes	8–45	10–30	10–10	11–45	12	14–15
linear, inch	5/16	3/8	3/8	29/64	29/64	17/32
Cut-off, in inches	19⅜	17⅜	20 7/16	18 9/16	20 5/16	18⅝
	.706		.75		.748	
decimal of stroke	.745	.668	.786	.714	.781	.716
Release in decimal of stroke	.916	.937	.925	.942	.921	.935
Exhaust lead, in inches	1 33/64	1 1/32	1 29/64	61/64	1 31/64	1 5/64
Compression, in inches	3¼	3⅝	2⅞	3¼	3 1/16	3¼
decimal of stroke	.125	.139	.110	.125	.118	.125
Steam opening, inches	1 9/32	1 23/64	1 27/64	1½	1 29/64	1 35/64
Exhaust opening, inches	1½	1½	1¾	1¾	2–2	2–2
	4,316		5,600		7,635	
Velocity steam, feet per minute; piston speed, 541.66 feet per minute	4,445	4,187	5,750	5,450	7,870	7,400
Velocity exhaust, feet per minute; piston speed, 541.66 feet per minute	3,800		4,670		5,725	
Velocity of steam through throttle	5,560 feet per minute.........5-inch diameter.					
exhaust to condenser	4,660 feet per minute.........15-inch diameter.					

Crank Shaft.—The crank shaft is of the built-up type, having the shafts and pins keyed and shrunk on the crank arms. There are axial holes through shaft and crank pins. The cranks are 120 degrees apart.

Shaft Data.

Crank shaft, total length, feet and inches	13–8¼
diameter, inches	8½
axial hole, diameter, inches	3½
coupling disc (1), diameter, inches	17
thickness, inches	2½
bolts, number in coupling	6
(taper), diameter at face of flange	2$\frac{3}{16}$
journals, diameter, inches	8½
length, inches	9¼
crank pins, diameter, inches	8½
length, inches	9⅜
axial hole, diameter, inches	3½
webs, width, inches	16½
thickness, H.P., inches	5¼
I.P., inches	6¼
L.P., inches	7
Thrust shaft, total length, feet and inches	8–0
diameter, inches	8½
diameter axial hole, inches	3½
collars, number	6
diameter, outside, inches	14
inside, inches	8½
thickness, inches	1¼
space between, inches	4
Line shaft, number	2
length each, feet and inches	19–0
diameter, inches	8
diameter axial hole, inches	3½
Propeller shaft, length, feet and inches	21–5½
diameter, inches	9
diameter axial hole, inches	3½
length, taper for propeller hub, inches	18¼
taper, inches diameter per foot length	1
axial hole through taper, diameter, inches	2
casing at bearings, thickness, inch	0⅝
brass sleeve through stern tube, thickness inch	0⅛
Couplings, diameter, inches	17
thickness, inches	2½
bolts to each	6
pitch circle, diameter, inches	13

Thrust Bearing.—The thrust bearing is of the usual horseshoe type, of cast iron, with a steady bearing at each end. It rests on a cast-iron sole plate, securely bolted to the foundations and is adjustable, in a fore-and-aft direction by means of wedges, fitted between the ends of the bearing and lips, cast on the sole plate. There are six cast-iron horseshoes, faced with white metal which is grooved for the proper distribution of oil, and the horseshoes are adjustable, by means of nuts on 1¾-inch-diameter side rods. The steady bearings have cast-iron bottom boxes and caps, both lined with white metal, and each cap is secured by two 1½-inch-diameter bolts. The body of the thrust bearing forms an oil well which is cooled by a water-service pipe, and there are stuffing boxes and glands, at the ends of the steady bearings, to retain the oil. Provision is made for circulating water through each horseshoe.

Spring Bearings.—There are three spring bearings of cast iron, lined with white metal. They are fitted, at the ends, with stuffing boxes packed with lamp wick.

Stern-Tube Bearings.—There is a forward and an after stern-tube bearing, connected by a cast-iron stern tube. Each end of stern tube is fitted with a composition sleeve, in which the lignum-vitae bearing proper is installed.

Bearing Data.

Main bearings, white metal lined, number	6
diameter, inches	8½
length, inches	9¼
Thrust bearing, white metal faced shoes, number	6
steady bearings, number	2
diameter, inches	$8\frac{1}{2}-\frac{5}{1000}$
length, inches	8
Line shaft (spring) bearings, white metal lined, number	3
diameter, inches	$8-\frac{1}{64}$
length, inches	16
Stern-tube bearings, lignum-vitae lined,	
forward bearing, diameter, inches	10¼
length, inches	15¾
after bearing, diameter, inches	10¼
length, inches	30½

PROPELLER.

There is one right-hand, four-bladed, manganese-bronze propeller with blades cast with hub. The blades are machined true to pitch and the hub has a taper fit on the shaft and is secured by one key and a nut. The taper of shaft is 1 inch in diameter per foot in length.

Propeller Data.

Diameter of propeller, feet and inches	9–08
hub, inches	22
Pitch, feet and inches	11–06
Ratio, diameter to pitch	00.84
Area, projected, square feet	29.24
helicoidal, square feet	36.32
disc, square feet	73.39
Ratio, projected to disc area	.398
helicoidal to disc area	.494
Distance, upper tip of blade from hull of ship, inches	16

CONDENSING APPARATUS.

Main Condenser.—There is one main condenser, cylindrical in form, located on the port side of the engine room. The circulating water passes through the tubes, entering at the bottom and passing out at the top. The tubes are straight, rolled into the tube sheet at the back and packed, with the usual glands, at the circulating water-entering end. The principal dimensions are as follows:

Inside diameter, feet and inches	3–06
Thickness of shell, steel, inches	00¼
Length between tube sheets, feet and inches	6–01
Thickness of tube sheets, inches	1 and 1¼
Tubes, number	1,113
diameter, outside, inches	00⅝
thickness, mils	65
Cooling surface, square feet	1,107
Main exhaust opening, diameter, inches	15
Auxiliary and dynamo-exhaust opening, inches	6
Air-pump suction, diameter, inches	4½
Circulation-water inlet and outlet, diameter, inches	8
Relief valve, diameter, inches	3½
Heads, cast iron, thickness, inches	0¾
Test pressure, shell, pounds per square inch	30

Main Air Pump.—The condenser is provided with a Blake vertical twin-bucket beam air pump with one steam cylinder 8 inches diameter, two water cylinders 16 inches diameter each and a common stroke of 12 inches. The suction and discharge pipes are 6 inches and 5 inches diameter respectively.

Main Circulating Pump and Engine.—There is one double-inlet, centrifugal circulating pump, operated by a vertical. single-cylinder engine. The crank shaft of the engine is forged-steel, Class A, and the impeller shaft is manganese bronze. The crank shaft is supported by two composition bearings lined with white metal and the impeller shaft by two bearings lined with lignum-vitae. The engine is enclosed, to just below the cylinder, to prevent the escape of oil from the forced-lubrication system. The forced-lubrication pump is operated by an eccentric on crank shaft, located between the flywheel and adjacent bearing. The main features of pump and engine, are as follows:

Circulating-pump engine, number of cylinders	1
cylinder, diameter, inches	5
stroke, inches	6
impeller, diameter, inches	28
width at tip inches	1¼
type, inclosed.	
Suction opening, diameter, inches	8
Discharge opening, diameter, inches	8

Feed and Filter Tank.—There is a feed and filter tank, of combined capacity of 450 gallons, located on the port side of the engine room. The tank is of Class C steel, rectangular in shape, 3 feet 6 inches long, 3 feet 5 inches wide, and 5 feet 1 inch high. The tank is thoroughly galvanized and covered with 1½-inch non-conducting material which is lagged with galvanized sheet iron, 14 mils thick. The connections are:

One main air-pump discharge, diameter, inches	5
Three vapor-pipe connections, diameter, inches	1½
One main feed-pump suction, diameter, inches	3
One auxiliary feed-pump suction, diameter, inches	3
One drain from heating system and whistle, diameter, inches	1¼

One drain from evaporator steam traps, diameter, inches	1
One feed-heater drain, diameter, inches	1½
One drain from scuttle-butt, diameter, inches	1
One overflow connection, diameter, inches	3

ENGINE–ROOM AUXILIARIES.

Auxiliary Condenser.—There is one auxiliary condenser, fitted with a turbo-circulating pump, and an independent air pump, located on the starboard side of the engine room. The condenser is cylindrical, with a steel shell, composition tube sheets and cast-iron water heads. The tubes are expanded into the tube sheets at the circulating-water end and are packed, with the ordinary type of gland, at the back. General particulars are as follows:

Shell, diameter inside, inches	24
thickness, inches	$0\frac{3}{16}$
Thickness tube sheet, front end, inches	1⅛
back end, inches	0⅞
Length between tube sheets, feet and inches	3–6½
Tubes, number	384
outside diameter, inches	0⅝
thickness, mils	65
Cooling surface, square feet	222
Circulating-water connections, diameter, inches	4
Exhaust connection, diameter, inches	7
Test pressure, pounds per square inch	30

Auxiliary Air Pump.—There is one auxiliary air pump; Blake, vertical, simplex, featherweight, 4½ by 8 by 6 inches, having a 3½-inch-diameter suction and a 3-inch-diameter discharge, for use with the auxiliary condenser.

Auxiliary Circulating Pump.—There is one auxiliary circulating pump. It is a Worthington, horizontal, volute pump with an impeller 8 inches in diameter. The impeller has 7 vanes. Diameter of suction, 5 inches; discharge, 3 inches. The pump and impeller are of composition "G."

STEAM, EXHAUST AND FEED PIPES.

Main Steam Pipes.—A 3½-inch-diameter main steam pipe leads from the stop valve on each boiler and, uniting in the

boiler compartment into a 5-inch-diameter pipe, passes through the after fireroom bulkhead, with a stop valve on the engine-room side of the bulkhead, and leads to the main engine throttle valve. There is no main steam separator.

Auxiliary Steam Pipe.—At each main boiler stop valve there is a 3-inch-diameter auxiliary steam stop valve from which a pipe leads aft, on each side of the boiler compartment, passes through the engine-room bulkhead, and supplies the auxiliaries on its side of the ship. In the boiler room the auxiliary steam pipes are cross-connected forward of the boilers, and from this cross-connection a 2-inch-diameter branch supplies steam to the forward auxiliaries.

Auxiliary Exhaust Pipes.—There is the usual auxiliary exhaust system with connections to the main and auxiliary condensers, feed-water heater, and the atmosphere. The dynamo turbines can also exhaust to the atmosphere.

Feed and Fresh-Water Systems.—The main feed pump, located in the engine room, draws from the feed tank, the condenser, and the reserve feed-water tanks. It discharges through, or by-passes, the feed-water heater and leads to the main feed-stop and check valves on the boilers. The auxiliary feed pump, in the boiler room, draws from the feed tank and the starboard and port reserve feed-water tanks, discharging to the auxiliary feed-stop and check valves on the boilers and to the reserve feed-water tanks. All feed-discharge pipes are fitted with air chambers at the boilers.

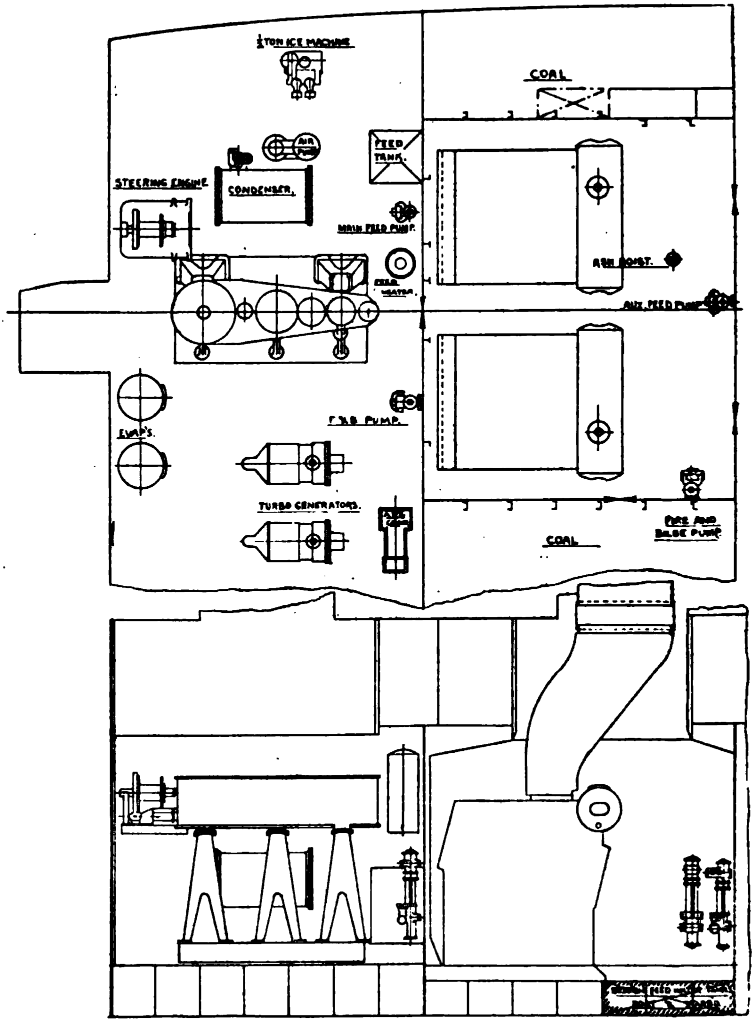
BOILERS.

There are two Babcock and Wilcox water-tube coal-burning boilers, in a common watertight compartment, arranged as shown. The boilers are designed to supply the necessary steam for operating all machinery, at full power, under natural draft. They are fitted with inswinging furnace and ash-pit doors.

The uptakes are of the usual design. There is one smoke-

U. S. S. Sacramento.—Pumps and Pump Connections.

No.	Pumps.	Size (inches).	Kind.	Suctions from— Ins.		Discharge to— Ins.		Location.
1	Main air.........	8 × 16 × 12..	Twin vertical......	6	Condenser	5	Feed tank	Engine room.
1	Main circulating	28″ runner ... 5 × 6 engine.	Vertical single..... Centrifugal.	8 7	Sea....................... Drainage.	8	Condenser .... Water service.	Engine room.
1	Main feed	8 × 5 × 12 ...	Vertical piston..... D.A. single.	3 3	Feed tank............. Channel-way.	2	Main feed discharge.	Engine room.
1	Auxiliary feed ..	8 × 5 × 12 ...	Vertical piston..... D.A. single.	3 2½ 2½ 1½	Feed tank............. Ship's side. Reserve feed.......... Hose connection	2½ 1½ 2½	Auxiliary feed discharge. Hose connection. Reserve feed.	Boiler room.
2	Fire and bilge....	8 × 7 × 12....	Vertical piston..... D.A. single.	4 4 1½ 2½	Sea....................... C. and R. drainage.. Hose conn. F.F...... Hose conn. E.R......	3½ 3½ 1½ 3½	Fire main.......... Overboard.......... Hose connection. Distillers.	1 engine room. 1 boiler room.
1	Auxiliary air. ...	4½ × 8 × 6 ...	Vertical single.....	3½	Auxiliary condenser	3	Feed tank	Engine room.
1	Auxiliary circulating.	8-in, runner, Terry turb,	Vertical single, centrifugal.	5	Sea	4	Aux. condenser ...	Engine room.
1	Lubricating oil..	3½ × 1¾ × 4...	Vertical piston, D.A. single.	¾	Storage tanks.........	¾	Distributing tank.	Engine room.
1	Evap. feed.......	3½ × 4 × 4....	Vertical piston.... D.A. single.	2 2	Sea .... Overboard discharge from distillers.	1½	Evaporators through feed heaters.	Engine room.
1	Distiller fresh water.	3½ × 4 × 4 ...	Vertical piston, D.A. single.	2	Bottom of distillers. Test tank.	1½ 1½ 1½	Fresh water tanks. Main feed tank. Reserve feed tanks	Engine room.



U.S.S.SACRAMENTO.

pipe, 76 feet in height above the base line and oval in section, the outside length being 7 feet 5 inches, and the outside width 5 feet 8 inches.

Boiler Data.

Number of boilers	2
Working pressure, pounds per square inch, by gage	215
Test pressure, pounds per square inch, by gage	325
Height to top, external, feet and inches	10–11¼
center of drum, feet and inches	10–04¼
Length on floor, feet and inches	10–01
Width on floor, feet and inches	9–00½
Drum, inside diameter, inches	36
length, feet and inches	9–06¼
thickness, inch	00½
Number furnaces, each boiler	1
furnace doors, each boiler	2
Length of grate, feet and inches	7–00
Width of grate, feet	7.71
Grate surface, square feet, one boiler	54
total, square feet	108
Heating surface, one boiler, square feet	1,908
total, square feet	3,816
Ratio, heating surface to grate surface	35.33
Number of headers, each boiler	15
2-inch tubes, each boiler	350
4-inch tubes, each boiler	15
Exposed length of 2-inch tubes, feet	9
Area of smoke pipe, square feet	15.55
Ratio grate surface to smoke-pipe area	6.94
Diameter, main steam, boiler stop valve, inches	3½
auxiliary steam stop valve at boiler, inches	3
main and auxiliary feed stop and check valves, inches	2
surface blow valve, inches	1½
bottom blow valves (two), inches	1½
safety valve, duplex, inches	3

EVAPORATING AND DISTILLING APPARATUS.

The evaporating and distilling plant is located in the after end of the engine room. There are two vertical evaporators, arranged to operate in single or double effect; two distillers in the engine-room hatch; an evaporator feed-water heater; an evaporator feed-pump, distiller fresh-water pump, and all accessories. The plant has a combined normal capacity of 5,000 gallons of fresh water per twenty-four hours and, when clean, an overload capacity 40 per cent. in excess of the normal capacity. Circulating water for the distillers is supplied

by the engine-room fire and bilge pump, and there is an emergency connection from the fire main.

Evaporator Data (*each*).

Type and number	2 No. 12, Reilly, vertical, multi-coil Navy Type
Diameter, inside, inches	36
Length over all, feet and inches	6–06
Coils, number	12
Heating surface, square feet	73.80
Steam connection, diameter, inches	2
Vapor connection, diameter, inches	2½
Feed connection, diameter, inches	1
Blow-off connection, diameter, inches	2½

Distiller Data (*each*).

Type and number	2 Reilly, vertical, multi-coil, Navy Type.
Diameter, inside, inches	24
Length over all, feet	5–0
Coils, number	8
diameter, inches	1
thickness, inches	.065
Cooling surface, square feet	49.2
Circulating-water connections, diameter, inches	2½
Vapor inlet, diameter, inches	2½
Drain, diameter, inches	0¾

Evaporator Feed-Water Heater.—There is one cylindrical evaporator feed-water heater of the bent-tube type. The shell is of copper, tubes Admiralty metal, tube sheets rolled Naval brass, and head of composition G. The particulars are as follows:

Heating surface, square feet	14.24
Inside diameter of shell, inches	9.5
Thickness of shell, inch	0.125
Length extreme, feet and inches	3–8.625
Number of bent tubes	9
Tubes, outside diameter, inches	2
thickness, mils	72
Inlet and outlet, diameter, inches	1½
Vapor connections, diameter, inches	3
Tube sheet, diameter, inches	13¼
thickness, inch	0⅞
Test pressure, pounds per square inch	50

WORKSHOP.

The machine shop, suitably equipped for this class of vessel, is located on the orlop deck, off the engine room, on the starboard side, between frames Nos. 41 and 48. Machine tools, each driven by an independent, inclosed, variable-speed, electric motor, and all with the most modern attachments and necessary tools and appliances, are installed as follows:

No.	Description.	H.P. of motors.
1	16¾-inch by 32⅜-inch swing, extension-gap lathe, 4 feet between centers	2 H.P.
1	Column, tool-room shaper, 15-inch stroke by 15-inch traverse	1 H.P.
1	Upright drill, to drill up to 1½-inch diameter and to at least 10½ inches from edge of work, spindle fitted for No. 4 Morse taper, with reducers, sockets, etc.	2 H.P.
1	Double emery grinder, on column, with attachment for surface grinding.	
2	Machinists' swivel bottom vises.	
1	Steel, blacksmith forge, portable folding type.	

ELECTRIC PLANT.

The electric plant consists of two 25-kilowatt generating sets, located in the starboard side of the engine room. Each generating set consists of a direct-current General Electric generator, compound wound, furnishing a pressure of 125 volts. The generator is direct connected to a horizontal Curtis steam turbine of the single-stage condensing or non-condensing type, with three rows of buckets mounted on the same wheel. A row of intermediate or stationary buckets are assembled between each two rows of revolving buckets. The nozzles are contained in a separate casting and are not adjustable. Steam is admitted through a strainer in the emergency-valve chamber to the steam chest and is passed through double-balanced poppet valves, operated by the governor, direct to the steam nozzles, thence through the turbine to the exhaust. For condensing operation the turbines are, in addition, equipped with a hand-operating valve controlling steam to supplemental nozzles, which may be used to increase the

capacity of the machine when operating non-condensing, or with low pressure or vacuum. Supplemental nozzles are controlled by the governor. The turbines are designed to operate under a steam pressure of 200 pounds per square inch and 25 inches of vacuum when exhausting to the condenser. The sets operate, normally, at 4,500 r.p.m. Both main and outboard bearings are forced-feed lubricated, from a geared pump, driven from a worm and gear on the outboard end of the generator shaft. In emergency the outfit can be operated without forced lubrication. All lighting feeders are energized from the switchboard, located in the starboard corner of the engine room, at frame No. 46. In addition to supplying 300 fixtures installed for lighting the vessel, truck lights, running lights, night signalling set, one 18-inch hand-controlled searchlight and interior communication, current is furnished for wireless outfit, ventilating motors, potato peeler, fresh-water pump motor, sanitary-pump motor, workshop motors and deck-winch motor.

REFRIGERATING PLANT.

There is a ½-ton vertical, steam driven, Allen dense-air ice machine located in the engine room on the port side. The cooling pipes from the machine are led to the ice tank, scuttle butt and the cold-storage room. The ice machine is provided with a recooler. The cold-storage room, located on the berth deck between frames Nos. 60 and 63, on the port side, is subdivided into meat room, crew's refrigerating room and officers' refrigerating room. The capacities of these compartments are respectively, 293, 125 and 74 cubic feet, while the ratios of capacities to cooling surfaces are respectively 2.1, 2.25 and 2.1. The piping is arranged for the usual cut-outs, etc. A connection is provided for furnishing circulating water from the sanitary-pump discharge. There is a standard 4-can ice-making box, close to the ice machine. The scuttle butt is located on the berth deck between frames Nos. 51 and 52.

TRIALS.

The trials of the *Sacramento* were run over the course off the Delaware Breakwater, March 31 and April 1, 1914. The standardization trials were run first, and it was decided to run these trials as follows:

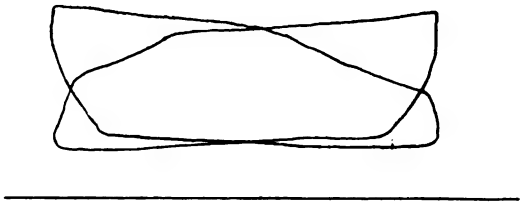
Three runs at 8 knots.
Three runs at 10 knots.
Three runs at 12 knots.
Four runs at highest speed attainable.

The standardization runs were begun at 9:39 A. M., March 31, 1914, and were completed at 1:30 P. M. the same day. The conditions were fair, with a gentle to moderate breeze from E. ½ S. with choppy sea from same direction, but not enough sea to cause the ship to roll. The results of the standardization runs were as follows:

No of runs.	Speed, knots.	R.P.M.	I.H.P.
1	6.923	74.14	195
2	8.511	76.74	216
3	6.693	75.11	213
4	10.946	97.44	427
5	8.918	98.34	449
6	11.029	99.51	445
7	11.465	121.00	838
8	12.645	122.88	860
9	11.830	121.15	857
10	13.230	133.24	1,129
11	13.260	134.27	1,164
12	12.574	129.67	994
13	13.148	130.49	1,027
14	12.380	130.71	1,031

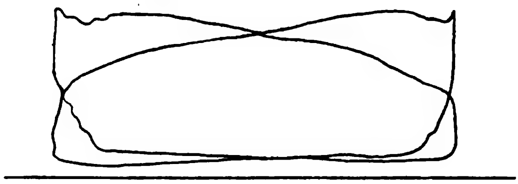
H.P.CYLINDER

M.E.P............ 95.5
M.R.P............ 14.77
I.H.P............317.
Scale of spring..120



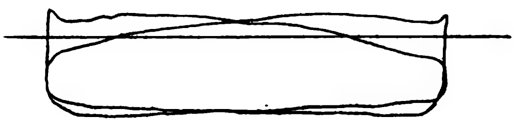
I.P.CYLINDER.

M.E.P............ 38.5
M.R.P............ 14.06
I.H.P............363.
Scale of spring.. 40



L.P.CYLINDER.

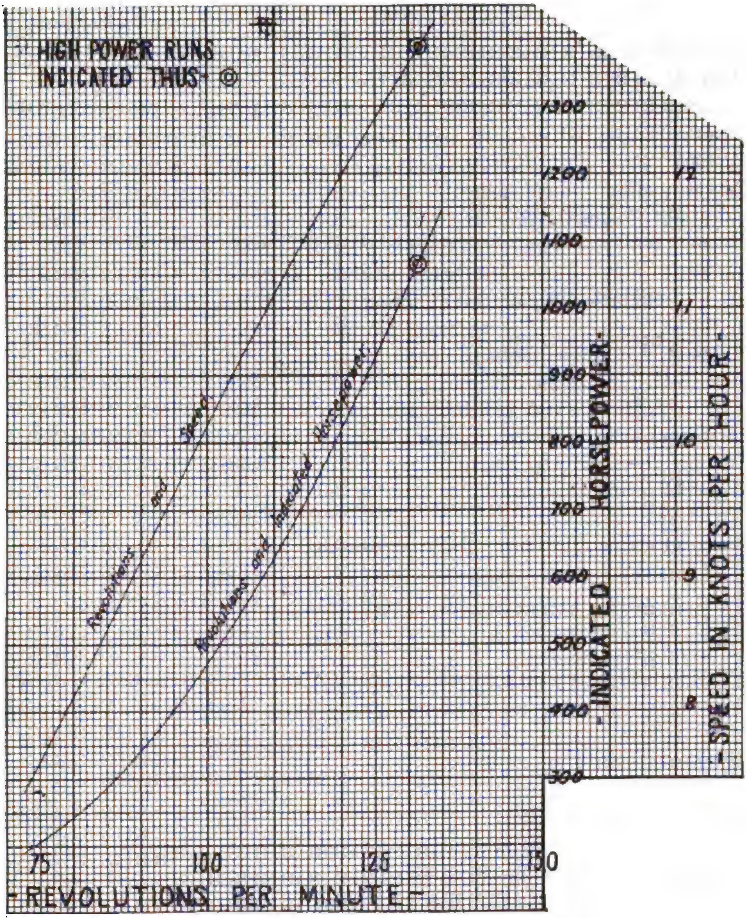
M.E.P............ 14.6
M.R.P............ 14.60
I.H.P............376
Scale of spring.. 20



U.S.S.SACRAMENTO.
FOUR HOUR FULL POWER TRIAL

Revolutions per minute, 129.75

Total I.H.P........... 1056
" M.R.P........... 43.43
Card No.3. 3/31/14



U. S. S. "SACRAMENTO."—STANDARDIZATION TRIALS, MARCH 31, 1914.

Data of 4–Hour Full-Power Trial and 10–Hour 10–Knot Trial.

	4-hour full-power trial. (*b*)	10-hour 10-knot trial. (*c*)
Date of trial	March 31, 1914	March 31, April 1, 1914.
Speed, per hour, knots	12.781	10.077
Mean draught at beginning of trial, feet and inches	11–3⅝	11–3⅝
Mean draught at end of trial, feet and inches	11–2⅞	11–4½
Corresponding displacement at mean draught during trial, tons	1,395	1,392.3
Steam at boilers, per gage, maximum, pounds	210	205
average, pounds	204.4	200.6
engines, per gage, maximum pounds	210	210
average, pounds	204	201.9
Steam at H.P. valve chest, per gage, pounds	192.3	126.
I.P. received, absolute, pounds	67.4	40.4
L.P. received, absolute, pounds	20.3	11.8
Maximum average revolutions per minute	130.61	100.57
Average revolutions per minute	129.50	99.71
Mean effective pressure, lbs. per sq. in., H.P. cyl	92.88	61.2
Mean effective pressure, lbs. per sq. in., I.P. cyl	38.89	21.95
Mean effective pressure, lbs. per sq. in., L.P. cyl	13.72	6.89
Referred pressure, lbs	42.14	24.39
Cut-off, decimal of stroke, H.P. cylinder	.664	.572
I.P. cyl	.750	.710
L.P. cyl	.748	.710
Vacuum in inches, maximum	28.8	28.9
average	28.56	28.23
Barometer in inches	29.98	30.00
Auxiliaries, double strokes, main air pump	31.4	25.3
Auxiliaries, r.p.m., main circ. pump	210.2	166.1
Auxiliaries, double strokes, main feed pump	23.2	17.2
Auxiliaries, r.p.m., sanitary pumps	1,508.	1,516.
dynamo steam pressure, pounds	198.	197.9
exhaust	66.3	24.2
ampères	180.1	178.2
Temperatures, injection, degrees	38.	37.3
discharge	70.8	63.5
air pump discharge	67.5	59.9

	4-hour full-power trial. (*b*)	10-hour 10-knot trial. (*c*)
Temperatures, feed	195.2	211.8
outside air	42.	42.
engine-room, working level	73.6	78.
fireroom, working level	61.3	79.9
I.H.P. high-pressure cylinder	307.	156.
intermediate-pressure cylinder	362.	159.
low-pressure cylinder	353.	136.
total	1,022.	451.
all machinery per square foot of grate surface	9.463	4.176
Square feet of heating surface per I.H.P. (total)	3.734	8.461
Pounds coal per I.H.P. per hour (main engines only)	1.761	2.395
Pounds coal per square foot of grate per hour	16.67	10.00
Kind and quality of coal	Pocahontas, run of the mine.	
Average pounds coal used per hour	1,800.00	1,080.00
Cooling surface, main condenser, sq. ft. per I.H.P., main engine only	1.08	2.454
Slip of propeller in per centum of its own speed	13.057	10.951

SALT-WATER EVAPORATORS.

CALCULATION OF BLOW-DOWN LOSS, HEAT BALANCE, ETC.

By WM. L. DEBAUFRE, MECHANICAL ENGINEER, ASSOCIATE.

While the blow-down loss in operating evaporators for the production of fresh water on shipboard is only a small proportion of the total heat energy imparted to the brine to produce evaporation, it is of interest to know how this loss varies with the method of operation. The effects of the rate of evaporation, of the quantity and concentration of the brine contained in the evaporator, of the method of discharging this brine, and of the degree of salinity of the feed, are all shown by the formula derived below. This formula was derived primarily to determine the relative blow-down losses of an evaporator with normal sea-water feed having a salinity of $\frac{1}{32}$ and with feed having a salinity of about one-fourth of this amount, as obtains with Severn River water used in evaporator tests at the Naval Engineering Experiment Station, Annapolis, Md.

Before proceeding further, it is necessary to settle upon a proper definition of salinity to serve as a basis for the derivation of the formula for blow-down loss. Salinity is therefore defined as the ratio of the weight of solid matter to the weight of pure water in a brine. Thus, a brine of salinity $\frac{1}{32}$ would be formed by adding 1 pound of salt to 32 pounds of pure water, making 33 pounds of brine. Salinity should not be confused with specific gravity. The specific gravity of the above brine will not be $1\frac{1}{32}$ because the volume of the 33 pounds of brine will be greater than the volume of the 32 pounds of pure water to which the 1 pound of salt is added; consequently, the density will be less than $1\frac{1}{32}$.

The following notation will be used as illustrated in Fig. 1:

Let w = weight in pounds of pure water contained in the brine in the evaporator,

k = mean salinity of brine in 32ds,

n = variation from mean value of salinity of brine in 32ds,

f = salinity of feed in 32ds, and

E = rate of evaporation of fresh water in pounds per hour.

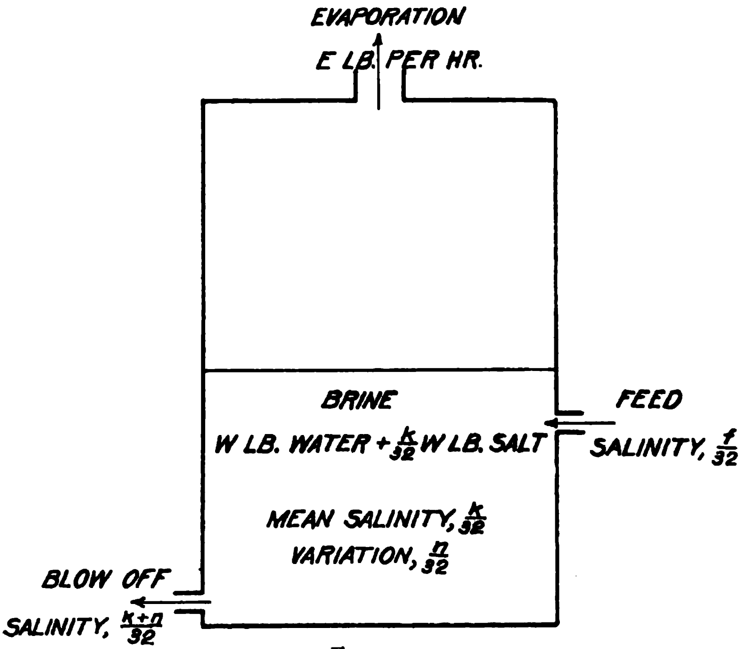


FIG. 1.

The maximum salinity of the brine in the evaporator reached just before blowing down will therefore be $\frac{k+n}{32}$. The minimum salinity just after blowing down will be $\frac{k-n}{32}$. During the period of evaporation the salinity will increase from

$$\frac{k-n}{32} \text{ to } \frac{k+n}{32}.$$

The above increase in salinity of $\frac{2n}{32}$ corresponds to an increase of $\frac{2n}{32}w$ pounds of salt to the w pounds of pure water in the brine present in the evaporator, which fact becomes evident by referring to the definition of salinity. This salt comes from the evaporation of a certain number of pounds of pure water which originally entered as feed of salinity $\frac{f}{32}$. That is,

$$\frac{\frac{2n}{32}w}{\frac{f}{32}} = \frac{2nw}{f} \text{ pounds} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

of vapor are formed during the evaporation period, which is consequently equal to

$$\frac{\frac{2nw}{f}}{E} = \frac{2nw}{fE} \text{ hours.} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

At the beginning of the blow-down period the salinity of the brine discharged is $\frac{k+n}{32}$. Under ideal conditions all the brine discharged would be of this salinity, while the remaining brine of the same density, containing say x pounds of pure water, would be diluted by feed of salinity $\frac{f}{32}$ containing say y pounds of pure water. The salinity at the completion of the blow-down period would be $\frac{k-n}{32}$. It is evident that,

$$x + y = w. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The amount of salt to the w pounds of pure water in the evaporator is $\frac{k-n}{32}w$ pounds at the completion of the blow down. This is made up of the salt in the brine of salinity

$\frac{k+n}{32}$ not blown out, $\frac{k+n}{32} \times x$ pounds, and of the salt in the make up feed, $\frac{f}{32} \times y$ pounds; that is,

$$\frac{k+n}{32} \times x + \frac{f}{32} y = \frac{k-n}{32} w. \quad . \quad . \quad . \quad . \qquad (4)$$

Combining equations (3) and (4), we obtain for the weight of pure water in the brine not blown out,

$$x = \frac{k-n-f}{k+n-f} w \text{ pounds.} \quad . \quad . \quad . \quad . \quad . \qquad (5)$$

Hence,

$$w - x = \frac{2n}{k+n-f} w \text{ pounds.} \quad . \quad . \quad . \quad . \qquad (6)$$

is the weight of pure water in the brine discharged at each blow down.

The weight of pure water contained in the brine blown out per hour of the evaporation period is evidently equal to (6) divided by (2); or,

$$\frac{\frac{2n}{k+n-f} w}{\frac{2nw}{fE}} = \frac{f}{k+n-f} E \text{ pounds.} \quad . \quad . \qquad (7)$$

Expression (7) shows that the weight of pure water in the brine discharged per hour (neglecting the time of blowing down) is independent of the amount of brine maintained in the evaporator (since w cancels out of numerator and denominator), is directly proportional to the rate of evaporation E, increases with higher degrees of salinity f in the feed but at a more rapid rate, decreases with a higher mean salinity k maintained in the evaporator brine, and is less the greater the variation n permitted in the brine density. The above facts are brought out in the curves, Fig. 2 to Fig. 5.

The curves in Fig. 2 show how the weight of pure water in

the brine discharged per pound of fresh water evaporated varies with the salinity of the feed for several ranges of the salinity of the brine within the evaporator from a mean value of 2 thirty-seconds. The curves in Figs. 3, 4 and 5 show the

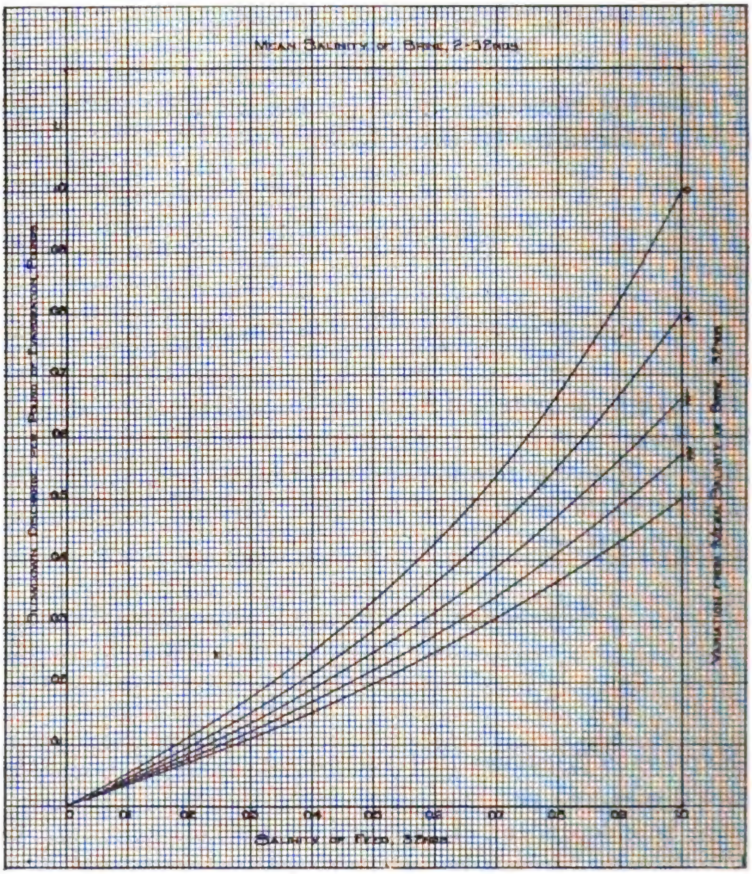


FIG. 2.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY BEING 2 THIRTY-SECONDS.

corresponding variation when the mean salinities are 2.5, 3, and 4, respectively. It will be noted that the greatest discharge occurs in each case when the variation in salinity is zero, that is, for operation with a continuous blow. The

apparent conclusion ought not be at once drawn, however, that evaporators should not be operated with continuous blow.

To further investigate the question of continuous *versus* intermittent blow, the curves in Fig. 6 were plotted. These

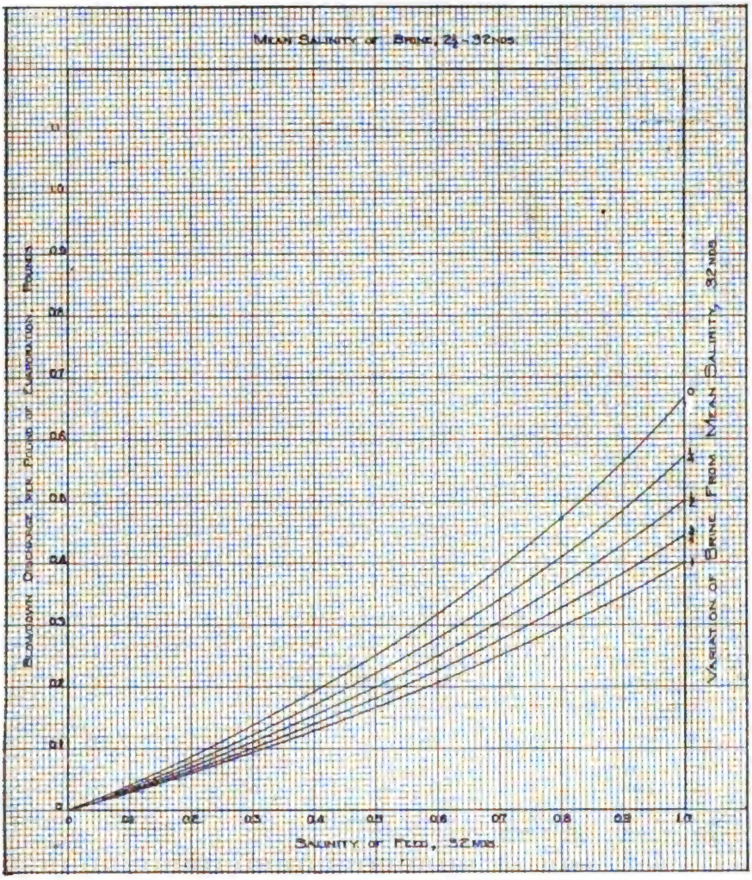


FIG. 3.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY OF THE BRINE BEING 2¼ THIRTY-SECONDS.

curves show the variation in the weight of pure water in the brine discharged per pound of fresh water evaporated, with the range of variation of salinity of the brine within the evaporator from the mean value and for several mean salinities,

the salinity of the feed being that of sea water, one thirty-second. The loss increases as the range of variation decreases, being greatest for no variation from the mean, corresponding to continuous blow. When no variation from the mean is

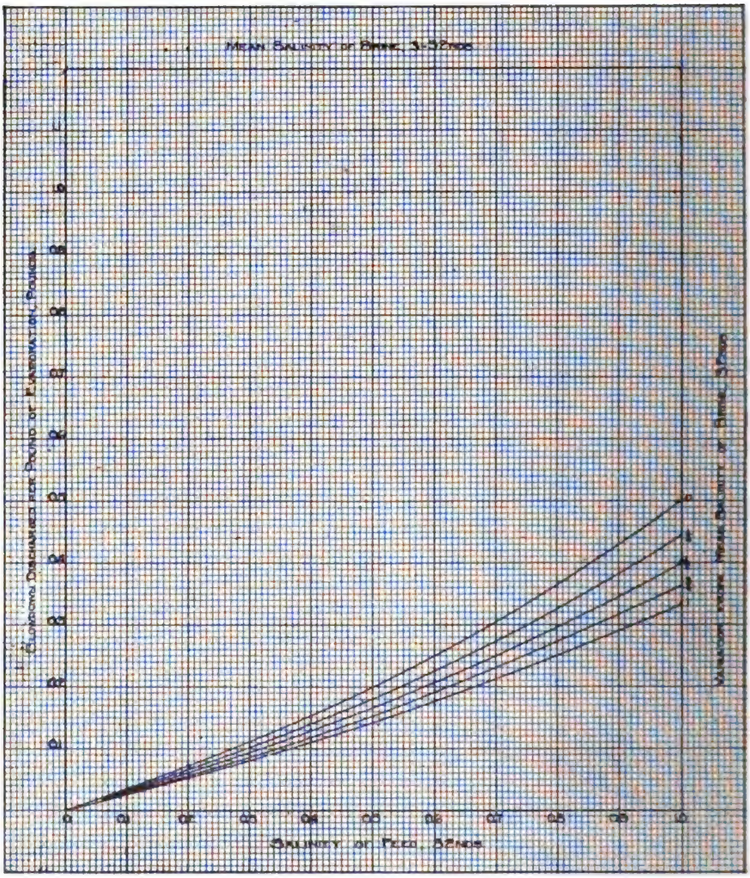


FIG. 4.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY OF THE BRINE BEING 3 THIRTY-SECONDS.

permitted, the maximum salinity evidently corresponds to the mean value; when variation is permitted, the maximum salinity is higher than the mean value by the range of the variation permitted from the mean value. It is now desired to

learn how the discharge varies with the range from the mean value for any given maximum salinity.

Assume a maximum salinity of 3 thirty-seconds; from Fig. 6 the discharge with continuous blow is 0.5 pound of pure

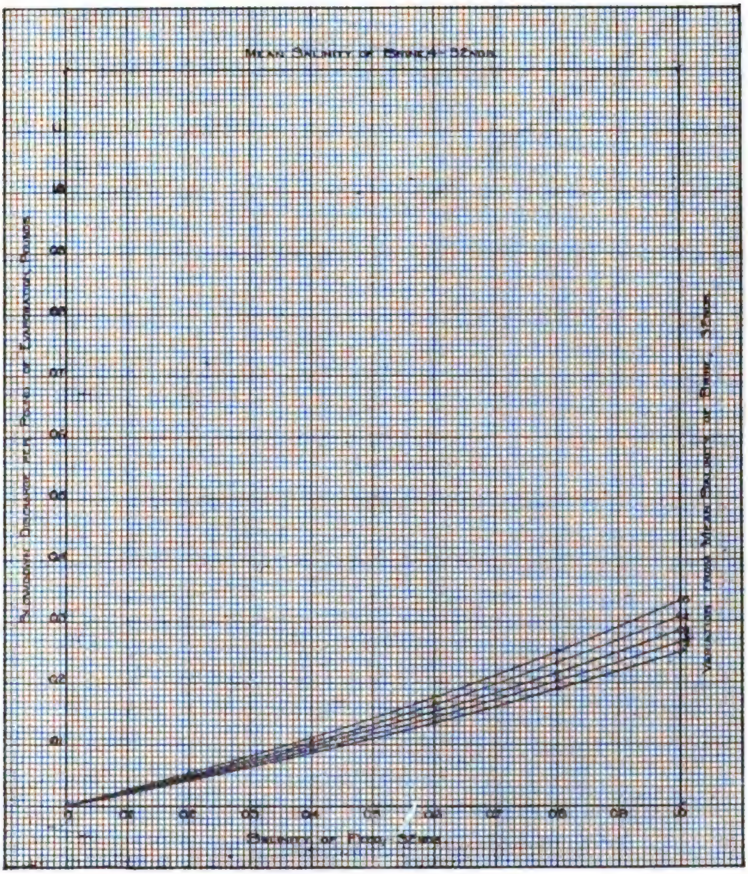


FIG. 5.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS RANGES IN SALINITY, THE MEAN SALINITY OF THE BRINE BEING 4 THIRTY-SECONDS.

water in the brine per pound of evaporation. If a range of 0.5 thirty-second from the mean is permitted, a maximum of 3 thirty-seconds corresponds to a mean salinity of 2.5 thirty-seconds; and for this case also Fig. 6 shows a discharge of

0.5 pound per pound of evaporation. The same discharge, as indicated by three curves, results from a maximum salinity of 3 thirty-seconds with any corresponding mean salinity and range. That is, the weight of pure water in the brine dis-

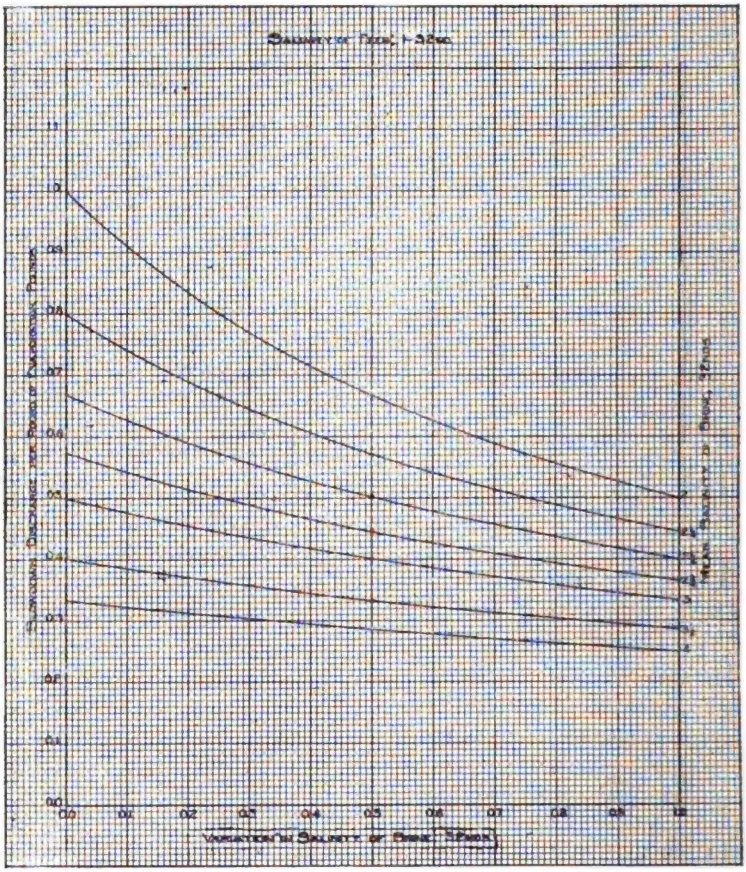


FIG. 6.—VARIATION OF BLOW-DOWN DISCHARGE WITH VARIATION OF SALINITY FROM MEAN VALUE FOR VARIOUS MEAN BRINE SALINITIES.

charge per pound of evaporation depends upon the maximum concentration attained in the evaporator and is independent of whether the blow down is continuous or intermittent.

The above fact might have been derived directly from

formula (7). Thus, instead of considering k and n as separate quantities, their sum might be replaced by m = maximum permitted salinity of brine in 32ds. Then, the weight of pure water in the brine discharged per pound of fresh water evaporated is equal to

$$M = \frac{f}{m - f} \text{ pounds. . . . . . . } \quad (8)$$

The curves in Fig. 7 and Fig. 8 are plotted for the variation of the weight of pure water in the brine discharged per pound of evaporation of fresh water with the maximum salinity m of the brine and with the salinity f of the feed, respectively.

The former, Fig. 7, shows the advisability of working at as high a maximum brine concentration as permissible, which concentration will be limited by the greater tendency of the evaporator to prime as the concentration increases. If the impurities in any part of the fresh water produced by the evaporator are limited to a certain number of grains per gallon, then it is immaterial whether the blow be intermittent or continuous, provided the maximum concentration is not exceeded at which the impurity of the fresh water is just within the desired limit.

If, however, the average impurity only is limited, or if the fresh water produced is used for two purposes in one of which a greater impurity is permissible than in the other, then the intermittent blow results in less loss. For, if the blow were continuous, the maximum concentration of the brine could not exceed that corresponding to the average impurity; while if the blow were intermittent, the maximum concentration might be higher because the greater impurity caused by the high concentration just before the blow down would be diluted by the purer water formed at the low concentration just after the blow down. Also, with intermittent blow, the fresh water produced just after blow down could be used for one purpose where very pure water is required, while the less pure water formed later could be used for the other purpose where less purity would be permissible.

The above conclusions are based upon the relative blow-down losses under the several conditions; but the total heat loss, as will be shown later, is a comparatively small proportion of the heat interchange involved. Consequently, these

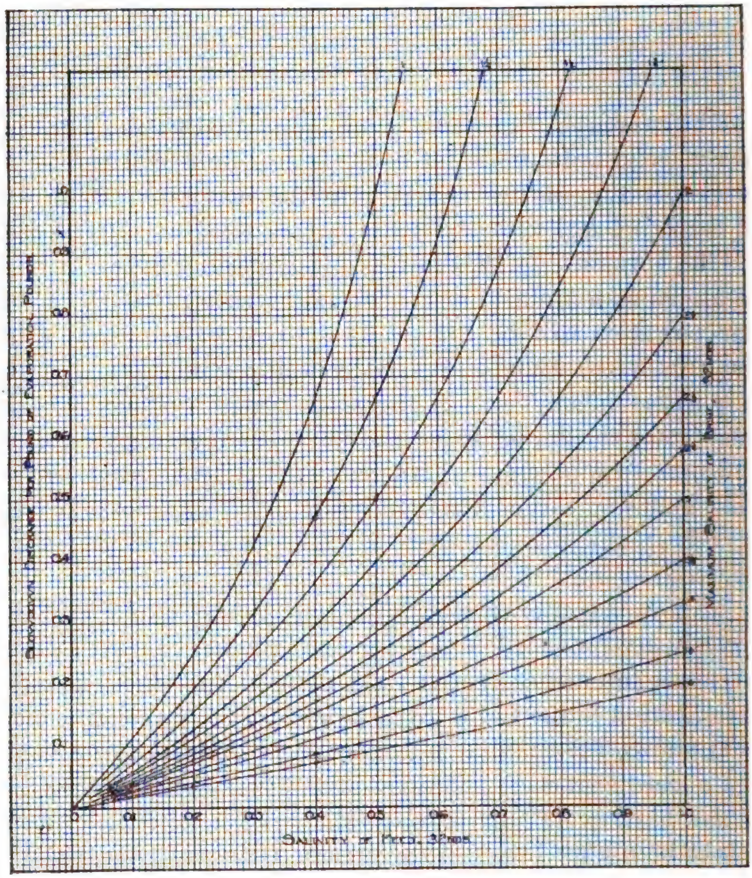


FIG. 7.—VARIATION OF BLOW-DOWN DISCHARGE WITH SALINITY OF FEED FOR VARIOUS BRINE CONCENTRATIONS.

conclusions may be modified by practical operating conditions. The main point is that the least loss occurs when the greatest maximum concentration of the brine considered allowable under the operating conditions is reached, and the loss is to

that extent independent of whether the blow be continuous or intermittent.

The curves in Fig. 8 are of greatest use when it is desired to calculate the blow-down discharge for a feed salinity different

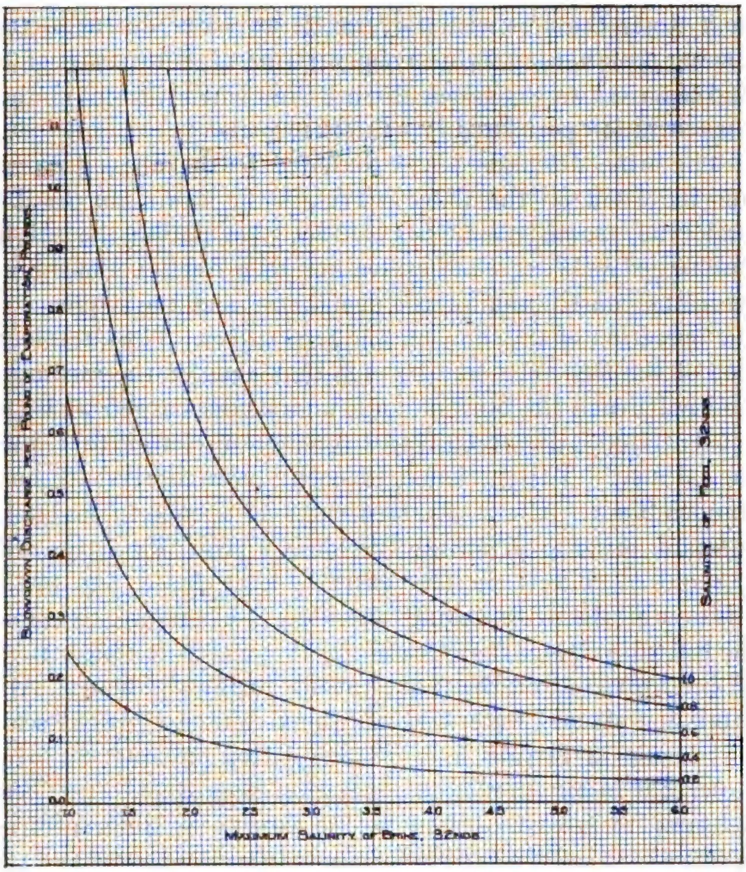


FIG. 8.—VARIATION OF BLOW-DOWN DISCHARGE WITH BRINE CONCENTRATION FOR VARIOUS SALINITIES OF FEED.

from that at which a test was made of an evaporator. It is evident that the feed salinity and the maximum brine concentration should be measured during the test. In the Severn River water available at the Naval Enginering Experiment

Station for testing evaporators there are from 200 to 350 grains of chlorine per gallon. At sea water of salinity 1 thirty-second contains about 1,200 grains per gallon, the salinity of Severn River water varies from 0.17 to 0.29 thirty-seconds, and the loss with sea water will consequently be from 5.9 to 10.8 times the loss with Severn River water during a test having a maximum brine concentration of 2 thirty-seconds, and from 4.7 to 8.3 times the test loss when the maximum concentration permitted is 3 thirty-seconds.

The brine discharged as calculated by the above formula and indicated on the above curves, is the minimum possible under the assumed conditions. In actual operation, the weight discharged will exceed these values by a fraction depending upon the construction of the evaporator and the sequence of the operation of the blow-off and feed valves. Thus, the location of the feed pipe may be such that some of the incoming feed will reach the blow-off pipe and be discharged with the concentrated brine, requiring a greater discharge to reduce the salinity. This is liable to occur particularly with continuous blow. Also if the feed valve is opened to maintain constant brine level during the blow, a larger discharge will be required than if the feed valve be kept closed, due to the dilution of the brine during the blow. Therefore, to obtain the least discharge, as indicated by the curves, the blow must be intermittent and the feed shut off during the blow. Any admission of cold water and additional blow to remove scale will result in further loss. The relative amounts of brine blown out in any method of operation should, however, bear the same ratio as indicated by the curves for different feed salinities so that the ratio determined from the formula or the curves can be applied to reduce the conditions with a measured feed salinity to those corresponding to a standard value.

The heat loss will equal the product of the weight of brine discharged, its specific heat, and some temperature difference, the upper limit of which will be the temperature of the brine discharged. This should nearly correspond to the vapor pressure maintained in the evaporator, a much less temperature

indicating dilution by cool feed. The fact that when the feed short circuits to the blow-off pipe, it is not heated to the concentrated brine temperature, shows that the blow-down loss from faulty construction is not as much as might be expected. Consequently, it is considered sufficiently accurate to calculate the weight by the formula proposed and assume the discharge temperature to be that corresponding to the vapor pressure as taken from steam tables, the temperature of boiling fresh water being slightly less than boiling brine for the same vapor pressure.

The correct lower temperature to assume will depend upon whether the efficiency of the evaporator alone or of the whole evaporating plant is under consideration. In the former case, the correct temperature will be that of the feed to the evaporator. In the latter case, it will be the temperature of the sea water available.

The values in the first two columns of Table I below are taken from "Marine Boiler Management and Construction" by C. E. Stromeyer.

TABLE I.—SPECIFIC HEAT OF SALT WATER.

Weight of salt added to 1 pound water, pound.	Specific heat of salt water, B.t.u. per pound brine.	Weight of brine containing 1 lb. pure water, lbs.	Specific heat of salt water, B.t.u. per lb. pure water.
1	2	3	4
.0	1.000	1.000	1.000
0.016	0.978	1.016	0.9936
0.049	0.945	1.049	0.9875
0.115	0.877	1.115	0.9779
0.123	0.871	1.123	0.9781
0.243	0.791	1.243	0.9832

The fourth column is the product of the second and third columns. The slight variation of the specific heat of salt water based upon one pound of pure fresh water contained therein indicates that no modification need be made of the foregoing conclusions based on the weight of pure water in the brine

discharged. For most calculations the value of this "specific heat" may be taken as unity.

In the heat balance of an evaporator considered apart from the complete water-distilling system, the only losses are the heat in the blow-down discharge and the heat radiated. The latter quantity is negligible with a well-lagged evaporator. The heat in the condensate from the steam supplied to evaporate the brine is not considered a loss when dealing with the evaporator alone, for this condensate is available as feed water to a boiler which then has to supply the latent heat only. All this latent heat is imparted to the brine to evaporate it. Consequently the above formula and curves will serve to calculate the efficiency in any given case. For example: Required the efficiency of an evaporator in which the shell pressure is maintained at 30 pounds per square inch gage, the salinity of the feed being 1 thirty-second and the maximum brine concentration 2.5 thirty-seconds. Assume a feed temperature of 70 degrees F. Now, corresponding to 30 pounds gage, the absolute pressure is 44.7 pounds per square inch, the corresponding steam temperature 274.0 degrees F., at which the latent heat is 928.6 B.t.u. per pound and the total heat is 1,171.5 B.t.u. per pound. The sensible heat at 70 degrees F. is 38.1 B.t.u. per pound. The formula or curves show that two-thirds pound of pure water will be contained in the brine blown out per pound of fresh water evaporated. There will thus be $\frac{2}{3} \times 1 \times (274 - 70) = 136.0$ B.t.u. in the brine discharged, and $1{,}171.5 - 38.1 = 1{,}133.4$ B.t.u. will be required to evaporate one pound of fresh water, necessitating a supply of $1{,}133.4 + 136 = 1{,}269.4$ B.t.u. Consequently, the efficiency of the evaporator will be $\frac{1{,}133.4}{1{,}269.4} = 0.8929$ or 89.29 per cent.

If the steam in the coils was maintained at 50 pounds gage during the evaporation, the ratio of water evaporated to steam condensed would be determined as follows: The latent heat of steam at 50 pounds gage, or 64.7 pounds per square inch absolute, is 911.2 B.t.u. per pound. Therefore, $\frac{911.2}{1269.4} = 0.7178$

pound of vapor is produced per pound of steam. It must be noted that the above calculations will be modified by moisture or superheat in the steam supplied and by superheat in the vapor produced. There should be no appreciable moisture in the vapor, for such would mean impure water.

Since the efficiency of an evaporator is so easily calculated under any assumed conditions, a test of a given evaporator is not conducted primarily to determine its efficiency but to determine its capacity and whether it will produce water of sufficient purity. Evaporator tests may also be conducted to determine the effects upon capacity and purity of various modifications in construction or operation. The efficiency might be affected by short circuiting some of the feed to the blow-off connection, but this effect should be slight because this short-circuited feed will not have been warmed to the evaporator temperature. In the above discussion, m could have been called the salinity of the discharge rather than the maximum salinity of the brine in the evaporator. This was avoided, however, to insure a definite understanding of the upper temperature to be used in calculating the heat lost in the discharge.

The upper temperature corresponds to the vapor pressure in the evaporator shell. Any feed short circuiting to the blow-off connection to increase the discharge does not increase the heat loss, assuming it to receive no heat while passing through the evaporator. The resulting calculations based upon this assumption are simpler than considering m to be the salinity of the discharge and then calculating the temperature resulting from the mixture of cool feed with the warm brine.

The formula and calculations have also been maintained simpler by dealing with the weights of pure water in the various brines rather than would have resulted by adding the weights of the saline matter to obtain the total weight of brine. If the total weight is desired, however, it can be easily obtained; thus, the total weight of brine discharged per pound of fresh water produced is $\frac{f}{m-f}\left(1+\frac{m}{32}\right)$.

SUMMARY.

The blow-down discharge of an evaporator may be calculated by the formula,

$$M = \frac{f}{m - f},$$

in which M = weight of pure water in the brine discharged per pound of fresh water evaporated, in pounds,

f = salinity of the feed, in 32ds, and

m = maximum salinity attained in the evaporator, in 32ds.

The salinity of the discharge will be m unless some of the feed short circuits to the blow-off connection and increases the amount of the discharge.

As the short-circuited feed will not with proper construction have sufficient time to absorb much heat, the heat lost in the discharge will equal,

$$H_m = M \times (t_v - t_f),$$

in which H_m = heat loss per pound of fresh water evaporated, in B.t.u.,

M = weight of pure water in the brine discharged per pound of fresh water evaporated, in pounds,

t_v = temperature corresponding to the vapor pressure in the shell, in degrees F., and

t_f = temperature of the feed, in degrees F.

The "specific heat" to raise the temperature of brine containing one pound of pure water one degree F., has been taken as unity.

The efficiency of the evaporator, neglecting the small loss by radiation, is given by,

$$e = \frac{H_v}{H_v + H_m},$$

in which H_m = value as calculated above, and

H_v = heat required to raise one pound of fresh water from the feed temperature to the temperature corresponding to the vapor pressure and evaporate it at that temperature, in B.t.u.

The above formulae indicate that the efficiency of an evaporator, considered apart from the whole distilling system, depends upon the salinity of the feed and the concentration attained in the evaporator. It is independent of whether the blow be continuous or intermittent, except in so far as this affects the short circuiting to the blow-off connection of feed which has been somewhat heated in passing through the evaporator.

Since the purity of the fresh water produced is dependent upon the average salinity of the brine in the evaporator while the loss in the blow down is dependent upon the maximum salinity, an intermittent blow down is a more economical method of operation than continuous blow. This is evident when we consider that the maximum salinity is equal to the mean salinity with continuous blow but is greater with intermittent blow.

The pounds of fresh water produced per pound of steam condensed is equal to

$$\frac{H}{H_v + H_m}$$

where H = heat given up by the steam in condensing, measured in B.t.u. per pound, and

H_v and H_m have the same significance as above.

DESCRIPTION AND TRIALS OF U. S. TORPEDO-BOAT DESTROYER *WINSLOW*.

BY LIEUTENANT W. F. COCHRANE, U. S. N., MEMBER.

The *Winslow* is one of six destroyers authorized by an Act of Congress approved August 22, 1912, these vessels being designated as Torpedo-Boat Destroyers Nos. 51 to 56, inclusive.

The contract for building three of these boats, Nos. 51, 52 and 53, named *O'Brien*, *Nicholson* and *Winslow*, was awarded to the Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Pa., and was signed December 7, 1912; the time allowed for completion being 23 months for the first boat, 23½ months for the second, and 24 months for the third.

The designed speed was 29 knots, at about 1,050 tons displacement.

The contract price for each vessel was $842,000.00, of which $502,000.00 was allotted for machinery.

The dimensions of vessel, machinery details, etc., are the same as those given in JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Volume XXVII, No. 2, of May, 1915.

TRIALS.

The contract required:

(*a*) A progressive trial over the measured-mile course at Lewes, Del., for standardizing the screws, extending from maximum speed (at least 29 knots) down to a speed of 8 knots.

(*b*) A full-speed trial of four hours' duration in the open sea in deep water, at the highest speed attainable, the average for the four hours not to be less than 29 knots. The speed to be determined by the average revolutions of the main shafts, according to the official standardization curve.

(*c*) A fuel-oil and water-consumption trial of four hours'

duration in open sea in deep water, at an average uniform speed of 24 knots, as nearly as possible. The trial to be conducted as nearly as possible to service cruising conditions.

(*d*) A fuel-oil and water-consumption trial of four hours' duration at 15½ knots, under conditions similar to the preceding trial, but with the cruising engines connected and in use.

(*e*) An endurance trial of ten hours' duration in the open sea at an average uniform speed of 15½ knots, as nearly as possible, following as closely as possible trial (*d*), with cruising engines connected and in use. Fuel oil and water consumption will not be measured on this trial, the purpose of which is to determine the reliability and endurance of the cruising engines.

(*f*) A fuel-oil and water-consumption trial of four hours' duration in the open sea, with the cruising engines connected and in use, at an average uniform speed of 12 knots, as nearly as possible.

(*g*) In addition to the above-enumerated trials the contract was amended to include a two-hours' trial at about 15½ knots, with the main turbines only in use, fuel oil and water consumption to be carefully measured on this trial.

Fuel-Oil Consumption Guarantees.—The contractors guaranteed that the fuel-oil consumption per knot run for all purposes, including that necessary for all auxiliaries in use on the trials, would not exceed 692.25 pounds at guaranteed maximum speed, 444.6 pounds at 24 knots, 213 pounds at 15½ knots, and 170 pounds at 12 knots, the consumption of the fuel oil at these speeds to be determined by the Trial Board from a curve based on the rate of fuel oil consumed on trials (*b*), (*c*), (*d*) and (*f*), and corrected to a standard of 19,500 B.t.u. per pound of fuel oil.

Standardization Trial (*a*).—This trial was conducted under favorable weather conditions on the measured mile at Lewes, Delaware, on June 29, 1915. Twenty-eight runs were made over the course at various speeds, and from the data obtained the revolution, speed and power curves in Plate I were plotted. Table I gives the standardization data.

From the curve in Plate I the following r.p.m. of the propellers were found to be necessary for the various speeds:

12 knots, ...	192.70
15½ knots, ...	249.20
24 knots, ...	413.40
29 knots, ...	571.20

Four-Hour 12-Knot Fuel-Oil and Water-Consumption Trial (*f*).—This trial commenced at 1:32 P. M., June 29, 1915, and was completed at 5:32 P. M., the same day. The weather was fair and the trial very successful. For data see Table II.

Four-Hour Full-Speed Trial (*b*).—This trial began at 7:55 A. M., June 30, 1915, off Five-Fathom Bank Lightship, and was completed at 11:55 the same day. The weather was fair. The data obtained on this run is given in Table II. The machinery operated excellently. All guarantees were easily attained.

TABLE I.—STANDARDIZATION U. S. S. "WINSLOW," LEWES, DEL., JUNE 29, 1915.

No. of run.	Speed, knots.	R.P.M. Stbd.	R.P.M. Port.	R.P.M. Ave.	S.H.P. Mean.	Speed Ave. Group.	S.H.P. Ave. Group.	R.P.M. Ave. Group.
1	8.46	131.49	130.29	130.89	234.00			
2	7.32	130.82	130.69	130.76	273.00	8.11	259	131.26
3	9.32	132.92	132.35	132.64	257.00			
4	11.39	200.77	199.99	200.38	797.00			
5	13.45	198.03	198.90	198.47	770.00	12.36	782	198.44
6	11.16	195.68	197.20	196.44	792.00			
7	17.41	253.49	254.07	253.78	1,540.00			
8	13.91	253.25	253.06	253.16	1,523.00	15.75	1,537	253.29
9	17.78	253.22	252.88	253.05	1,560.00			
10	18.33	330.81	330.58	330.70	3,453.00			
11	22.17	331.04	329.78	330.41	3,450.00	20.26	3,455	330.15
12	18.35	329.35	328.80	329.08	3,468.00			
13	25.82	419.74	416.54	418.14	7,566.00			
14	22.54	416.73	414.46	415.60	7,459.00	24.10	7,503	416.04
15	25.51	416.81	412.81	414.81	7,527.00			
16	24.67	470.07	467.19	468.63	10,763.00			
17	27.21	467.23	466.12	466.68	10,580.00	25.94	10,598	466.72
18	24.66	464.79	464.94	464.87	10,470.00			
19	29.73	565.83	562.61	564.22	15,904.00			
20	27.89	561.57	561.94	561.76	15,642.00	28.79	15,782	563.79
21	29.63	568.83	565.97	567.40	15,938.00			
22	29.01	594.53	594.48	594.51	17,558.00			
23	30.18	597.67	594.85	596.26	17,490.00			
24	29.53	598.85	604.14	601.50	17,915.00	29.85	17,828	600.86
25	30.23	607.94	603.65	605.80	18,221.00			
26	29.90	602.87	607.60	605.24	17,813.00			

Four-Hour 24-Knot Fuel-Oil and Water-Consumption Trial (*c*).—Following the full-speed trial the 24-knot trial began at 1:45 P. M., and was completed at 5:45 P. M. The weather was fair. The trial was very successful. For data see Table II.

Two-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (*g*).—The trial began at 6:30 P. M., following the 24-knot run, and was completed at 8:30 P. M. The trial was very successful. Data is given in Table II.

Four-Hour 15½-Knot Fuel-Oil and Water-Consumption Trial (*d*).—The trial began at 6:30 A. M., July 1, 1915, and was completed at 10:30 A. M., the same day. The trial was very successful. The weather was fair. For data see Table II.

Ten-Hour 15½-Knot Endurance Trial (*e*).—This trial began at 11:00 A. M., July 1, 1915, and was completed at 9.00 P. M., July 1, 1915. The weather was fair. The trial was very successful, and the reciprocating engines and clutches operated excellently.

TABLE II.—U. S. S. "WINSLOW."

......	4-hour full power.	4-hour 24-knot.	4-hour 15½-knot.	4-hour 12-knot.	2-hour 15½-knot.
Date of trial	6–30–15	6–30–15	7–1–15	6–29–15	6–30–15
Displacement	1,041	1,042	1,046	1,047	1,057.5
Boilers in use	4	4	2	2	2
Heating surface	21,600	21,600	10,800	10,800	10,800
Speed in knots	29.054	24.036	15.560	12.08	15.567
R.p.m., starboard	573.21	414.30	250.35	193.69	250.55
port	573.09	414.30	250.19	194.34	250.18
mean	573.15	414.00	250.27	194.02	250.37
S.H.P., mean	15,984	7,384.00	1,593.0	672	1,524
I.H.P., cruising engines	......	......	......	......	......
Pressures:					
Main steam (G)	243.69	234.69	246.0	225	238
Full speed (abs.)	S. 211.44	75.00	14.8	9.69	20
	P. 215.00	78.88	14.8	9.56	25
Cruising speed (abs.)	S. 157.94	207.25	40.63	23.13	65
	P. 157.18	201.13	34.63	20.00	65
10th stage (abs.)	S. 26.84	12.63	3.56	2.0	6.5
	P. 26.84	13.31	1.06	1.5	4.5
14th stage (abs.)	S. 14.0	6.8	1.93	1.31	2.5
	P. 13.25	6.0	1.50	1.0	2.0
Gland steam (G)	6.25	5.19	4.13	2.57	4.94
Cruising engines:					
H.P. valve chest	......	......	238.32	144	......
L.P. receiver	......	......	42.57	21.8	......
Vacuum, starboard	27.2	27.95	28.05	28.5	28.3
port	27.4	28.10	27.99	28.1	28.5
Auxiliary exhaust	8.87	8.38	7.5	3.6	6.25
Lub. oil to main engines	14.13	12.18	11.19	11.5	10.13
Oil to clutches	......	......	76.25	75.5	......
Temperatures:					
Air	74	73	80	74	70
Engine room	79	78	87	82	79
Auxiliary room	84	83	82	81	82
Fireroom	96	95	104	101	103
Oil to cooler	112	103	100	94	94
from cooler	90	58	90	86	86
Main injection	66	66	69	66	66
discharge	92	83	78	71	76
Fuel oil to heaters	73	74	88	......	78
from heaters	76	79	88	......	88
Smoke pipes	550	390	280	245	415
R.P.M. blowers	1,510	1,350 (2)	746 (1)	621 (1)	1,170 (1)
Air pressure	6.75	3.70	1.86	1.46	3.08
Water per S.H.P., M.E.	16.158	18.244	22.435	25.623	31.33
Water evap. per lb. oil	13.614	14.66	15.38	14.775	7.463
Fuel oil per knot run	661.50	388.0	138.0	105.0	199.0

OIL BURNING.

By A. M. R. Allen, Lieut., J. G., U. S. N., Member.

REFERENCES:

Journal American Society Naval Engineers,
Vol. XXIII, No. 2, "Oil Fuel," by Peabody.
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In taking up a new installation, the first thing to determine is the effectiveness of the pumps and heaters. Oil lines are tested to 600 pounds' pressure, and, in starting up, it is best to test everything, preferably with water, to see that all joints on lines and heaters, and all cocks and valves are tight. Try the line out in sections, taking care not to exceed this high pressure on any one section. Make notes of all leaks. Take particular care and see that the oil heater is tight. All modern heaters have outside packed joints, so that oil cannot get into the steam drainage. The Schutte-Koerting heaters have plugs so arranged that you can easily tell if there is any leakage from the steam to the oil side. Each installation has its necessary gages and test cocks arranged for just such testing, and, if not, it will be well to arrange for the installation of any test instruments and openings necessary in the very beginning, while still in port. The air flasks on the line, and all joints and valves connected with them, must be given particular attention.

After you have thoroughly tested and know that the oil line and valves are tight up to the burners, the next point to study is the burner spray. By testing with water and trying dif-

ferent pressures, one, or a certain range, will be found which will give the clearest spray. In doing this great care must be taken to see that the test burner, particularly the tip, is perfectly clean, or otherwise results will be very unreliable. The writer found that about 200 pounds pressure while steaming, and 150 pounds for port use gave very satisfactory results with the Schutte-Koerting system; while with the Ingram burner, using steam, about 40 pounds of oil pressure would give excellent results, as long as the burner was clean. As the burners get dirty, which shows by the increased pressure in the Ingram, and by streaks in the oil cone in the Schutte-Koerting and Bureau types, they must be removed and cleaned.

After a satisfactory spray pressure has been found, the next step is to so adjust the burner, if adjustment is possible, that the cone of oil just clears the brick rings in the furnace. These points are sometimes cared for in design, but in many cases, where Bureau burners have been substituted, the question of adjustment is entirely up to the engineer officer.

After the burners are right the next question is lighting off. Try out the hand pump and see what is the maximum pressure you can make with it. Open the line to the burners and stand by. In the meantime, see that the smoke-pipe cover is off, ventilator rigged, water at steaming level, auxiliary feed pump ready for use, and fuel-oil pump ready; air cock open, all gage glasses working, feed stops and checks working, bottom blows tightly closed, main and auxiliary stops closed, and safety valves and easing gear in working order. The last thing, look into the fire box through the registers and make sure no oil has collected in the furnace and that none of the burners are dripping. Prepare a long ⅜-inch iron rod, with a hook in one end to hold some waste for a torch, and put a little oil on it. Seeing the pressure on the line, light the torch and stick it in through the register, shutting the latter on it. Crack the burner valve, shutting off the valve if the oil does not ignite at once. This is the time for greatest care, for if an inexperienced man allows the furnace to fill with heavy

oil gas, which will not rise owing to the lack of draft, a heavy explosion will take place, injuring both the man and the boiler. It may be necessary, if in lighting off the first time the torch burns out before the burner lights properly, to get an air or steam hose from the deck to induce a good draft before again putting a lighted torch through the register. One point in particular is to be emphasized, and that is, that the man handling the torch must keep his face away from the register. The natural tendency is to get down close and see what sort of a stream the burner throws, but this must not be done. The only oil accident during the time I was engineer officer was due to a man doing this, and, fortunately for him, it only resulted in severe burns.

After the burner lights off properly the hand pump must be kept going until there is steam enough to run the pumps; then they must be started and the burners cut in as fast as possible, steam being also put into the heaters, if necessary. Oil, of course, becomes less viscous under heat, and flows more easily, and the point is to give just sufficient heat to make the oil have a viscosity of about 8 Engler and no more, for more simply wastes heat units.

Most destroyer installations will work well without a blower for port use, unless the evaporators are in use or an excessive amount of electricity is being used. This is best judged by the fireman from the appearance of his fire, and the engineer by the look of his stack. A light gray haze is the ideal condition, for the reason that it indicates complete combustion, giving a stack temperature of about 500 degrees on the pyrometer. Too little air causes excessive black smoke and panting, and too much air causes white smoke, which means that the excessive amount of air is combining with the small amount of oil and forming steam which is condensed when it reaches the air.

Panting, that condition of the boilers due to excessive amount of oil—more than the air supplied, can burn at the velocity at which it is being fed to the boiler, is the most in-

jurious factor in oil burning. It is a condition that should never be allowed to continue, for it shakes down the brickwork, loosens up the joints in the casing, and racks the boilers, connections, and ship itself in a highly injurious manner. It can be overcome by increasing the velocity of air, by slightly closing the registers, or by cutting out burners, or by increasing the air pressure by speeding up the blowers. Within limits the latter is the quickest and most effective measure, and the one chiefly employed in maneuvering. In this connection, it may be said that putting both blowers on a common throttle has tended greatly to increase the efficiency of a fireroom with two blowers, as it insures their being speeded up together, and in so doing prevents the increased pressure escaping through the slow burner, as was the condition before this was done. Panting is also frequently caused by insufficient air in the banks, which should always be charged as soon as there is steam on the ship. If air from the torpedo charger is used through a reducer, they can be charged at full working pressure, but if the Westinghouse compressor is used the oil pressure may have to be slightly lowered until through charging. The air banks, or "pigs" as they are called, should show a half a glass of air at working pressure, and must be pumped up frequently, as the oil absorbs the air. The pressure gage on the oil line should show a steady pressure regardless of the pump strokes, as no installation using mechanical atomization can be successfully operated unless this condition obtains.

The next test to be carried out is to test all casings, throat sheets and burner openings for tightness; also all openings into the firerooms themselves. This, of course, can best be done in a dead fireroom, and a good approximation for proper tightness is to close all openings, speed up the blowers to a fixed number of revolutions, and see what pressure can be maintained. All boilers must have some one inside to see where the air leaks into them, and all seams, stuffing boxes, doors, deadlights, etc., around the firerooms must be carefully gone over. If there appear to be any leaks, soap and water

will sometimes aid in detecting them. On the *Jarvis* the vent from C-2, the reserve-feed tank in the engine room, was open to the fireroom, and this caused an air loss and made false readings in the water tank. The blower fans should run with as little clearance as possible, otherwise there will be much leakage around them. In carrying out this test care should be taken to see that the air gage is accurately filled, and as leaks are stopped the increase of pressure noted. The point of all this is to make sure that all the air pressed into the firerooms passes out through the burner registers, and the economy lies in the fact that, for every leak stopped, the velocity necessary to keep a given number of burners going can be obtained for less revolutions of the blowers, which should be run with as little steam as possible. The fact must be borne in mind, that in free route, when sudden increase of power is not likely to be demanded, one blower run at full speed is more economical than two run slowly. To do this, however, the blower shut down must have an absolutely tight hood.

Another point to remember in blower operation is that, when using the common throttle care must be taken to adjust the independent throttle so that there will be an equal increase in speed in both blowers for a given opening of the common throttle. This is greatly facilitated by tail rods on the blowers, and deck throttles, so that a man on deck can adjust this. Once adjusted it should be frequently checked.

Testing of feed pumps, exhaust lines, feed heaters, etc., must be carried out, but these tests are not within the province of this article to consider.

The next test is a heat test, and the principle to be borne in mind is the fact that the least possible amount of steam that can be used to give the oil sufficient fluidity to burn properly is all that is required. This must be studied, as it varies with the quality of oil and the type of burner. The base of all experiments can be taken from the "Viscosity Table," gotten up at the oil-testing plant in Philadelphia, and only sufficient temperature used to give a viscosity of 8. The dis-

tance between the thermometer and the burners must also be considered, and it will take some little time to establish a set of standard conditions for any installation.

The three important tests having been completed, viz: pressure and temperature of oil, and air pressure, the next step is the establishment of standard working conditions for different speeds, and different boiler and fireroom combinations. The basis for this work should be, as far as possible, the work of the next previous destroyer from the same shipbuilding company, and the first efforts should be directed to proving out the previous results for your own installation. Unfortunately, up to the present time there has not been sufficient effort to put these results in such shape that new boats could take full advantage of the results obtained by others; and one trouble has been due to the lack of appreciation by younger engineers that only the most accurate data is of value. Too much care cannot be taken of air gages, pressure gages, thermometers and tachometers, and it must be constantly borne in mind that these instruments are the helpful sign boards that tell the engineer exactly what his installation is doing. The personnel must have this impressed upon them at all times, and must be led to realize that the care and watchfulness of the instruments on the oil system are just as important as the care of boiler gages and water glasses. They should be encouraged from the beginning to take an interest in keeping careful records of readings taken while on watch, and every effort made to show them the value of such readings. They will take interest themselves, as a rule, if you can show them clearly that the man who, while on watch, keeps everything carefully adjusted to standard conditions is the man who burns the least oil and who should get a reward for this careful work. Brains, not brawn, are required in an oil-burning fireroom.

Having established a system which will be productive of accurate data, it is the engineer's duty to so interpret this in-

formation that he can establish the *best possible standards* for all working conditions.

Underway the following method of burner adjustment must be used. The air pressure should be just sufficient to keep all burners required for a given speed going smoothly, without panting, and with a light gray haze showing from the stacks of all boilers in use. The center burners must be the first ones added, and a drop in steam pressure must be covered by immediately speeding up the blowers before cutting in more burners. Blowers must be immediately slowed when the demand for steam ceases, which is shown by the rapid rise of the gage pressure, but burners must be cut out first in this case. On several boats the engine-room annunciators have been connected to show the changes of speed in the firerooms, so that the W.T. can be prepared and handle his boiler quicker. Under ordinary working conditions the gage is the best indicator, and a properly-trained fireroom crew will hold the pressure without smoke and without popping off, except under the most unusual conditions. The W.T. has many things to watch, and if he is trained to watch his main steam gages, they are his greatest help.

One other element in training fireroom crews as separate watches and keeping them together like a gun's crew is that, for best economy, each boiler should have the same number of burners; and, when running two firerooms, this requires a good voice tube or some method of signalling between firerooms, otherwise one or the other will carry all the load instead of equally dividing it, as should be done under all conditions.

Governors on the fuel-oil pump have given general satisfaction, and they keep the oil pressure constant regardless of the number of burners in use. In maneuvering, the temperature will vary slightly due to the variation in the amount of oil passing through the heater, while the steam pressure remains constant, but this cannot be overcome in any of the installations so far installed.

Having once steadied on a given speed the standard for this speed is at once established as follows: The steam pressure must, of course, be kept constant at the highest pressure that can be carried without lifting the safety valves, and the usual method is to adjust the boilers so that each one is using the same number of burners. Then the air pressure is gradually cut down until a slight haze can be seen in the furnace through the peep, looking close under the tubes. The slightest inclination to pant is controlled by gradually closing the registers until the maximum opening for the conditions is found. The blowers are then kept steady at this speed. One boiler is assigned to carry the slight variations in steam pressure, while the other is kept constant except when necessary to clean a burner. This is shown by black streaks in the oil cone from the burner and by black smoke in the furnace. Well trained firemen will detect this at once, and, if there is a doubt as to which burner it is, will shift from one to the other until they find that one, which, when cut out, makes the furnace show clear again.

The reason that it is not possible to establish absolutely accurate standards for all fixed speeds is because of the variation in the power required to produce a fixed number of revolutions, due to change in conditions of wind, current, displacement, trim and the condition of the ship's bottom; but every effort should be made to find them for average conditions, in order to tell whether or not the speed is being made most economically.

As the revolutions are fixed, the number of burners in use is the best indication of economical working conditions; and if the number in use is excessive, investigation should at once be made to see that the vacuum, feed water, combination in use, etc., are such as to give best results. Usually, low temperature of feed water is the source of trouble. For instance, the 20-knot average working standards for the *Jarvis* are as follows: Main steam boilers, 260 pounds; engine room, 250 pounds; I.P.C., 155 pounds; M.H., 75 pounds; L.P.'s., 0;

Vacuum, 28; R.P.M., 465; back pressure, 4.5 inches; oil consumption, about 2,800 gallons a watch; and any drop in the feed temperature showed itself immediately in the number of burners required.

On ordinary runs the amount of make-up feed and oil used could be actually measured by soundings, and it was routine on all but the roughest trips to take them at the end of each watch.

The above notes will assist in getting a working basis for any installation, and the records kept should show the following:

Engine room.

R.P.M. No. 1.
R.P.M. No. 2.
R.P.M. No. 3.
Average R.P.M.
Speed.
Main steam (Boiler, E. R).
H.P.C.
I.P.C.
M.H.
S.L.P.
P.L.P.
S. Vac.
(Better the temperatures of the exhausts if thermometers could be installed).
P. Vac.
Back pressure.
Temperature highest bearing.
Stack temperatures.
Temperature hotwell.
Temperature air pumps discharges. $\begin{cases} S \\ P \end{cases}$

Firerooms.

Air pressure.
Oil pressure.
Oil temperature.
Feed temperature.
Number burners.
R.P.M. blowers.
Oil per watch gallons.
Make-up feed per watch.

Some of this data can be taken from the log, and some from a sheet or book kept by the W.T. on watch, and the whole compiled in card form for ready reference. This card record should be summarized at each visit to a navy yard, from which corrections should be made in the working conditions until the ship is down to most economical standards possible to at-

tain. The number of double strokes necessary for each pump, when working under best conditions, should also be known, as this will give a check on that auxiliary, in case a loss is noted which is hard to locate.

In shutting down, the main point to remember is to keep the blowers on until the demand for steam has been reduced below the point where a single burner can take care of that required; and, if this cannot be done, keep one blower running.

The care of the installation, as regards preservation, is well covered by the regulations. Fuel oil can be used to good advantage on the steel piping to keep it from rusting. The joints in the oil line are scraped flanges made up with shellac or Eretite or other joint preparation to stand 600 pounds' test pressure, and should be most carefully cared for. Fuel-oil pumps should be packed with leather rings and kept in the best adjustment at all times. Strainers must be cleaned frequently, and the slightest indication of loss of suction or decrease in pressure due to clogged strainer must be overcome at once by shifting strainers. Every oil-burning fireroom must have a vice for holding burners while cleaning them, and a proper equipment of wrenches and reamers. Great care must be taken of burner tips, as the slightest corrosion or carbon in the tips will give trouble. A steam connection to a burner holder for blowing out burners should be furnished near the vice, with a bucket held in a bracket underneath, so that it can be taken on deck and emptied, or, better, an arrangement made for blowing it through the side of the ship. Burner cocks or valves should always be kept tight, and the ship supplied with spare discs, or, if cocks are used, with the necessary reamers and blanks for renewing the cocks if necessary.

The registers and their operating gear should be kept thoroughly clean and in such condition that, when not in use, the shutters close the openings completely and tightly. The ship should have a large piece of canvas, which can be spread com-

pletely across the face of the boiler not in use, when steaming with one boiler in a fireroom. This will be held tightly over all openings by the air pressure, and prevent excessive loss through them.

The blower housings are designed to throw the air out over the boiler casings, and so warm it before it reaches the registers, but when operating in tropical climates this makes an excessive heat on the floor plates. To overcome this and preserve the health of the personnel it is usual to rig a wingless ventilator for each blower, which will bring a little of the fresh air down to the men on watch. In the *Jarvis* the temperature of the working platform was reduced from about 120–130 degrees to less than 110 degrees in this way. For port use a large winged ventilator will usually furnish sufficient air and make the fireroom livable, except in calm weather, when a blower must be used, not only for the boilers, but to prevent the men on watch being overcome by the heat.

The brick cones must be kept smooth and round, otherwise the projections will catch the spray and make the boiler smoke. The boiler bricking must be carefully examined after every run, and any tendency on the part of the brickwork to loosen up or fall down corrected at once by repairing the wall, using new bricks and high-temperature cement when necessary, or setting up on the fastenings of the old ones, the whole being thoroughly washed over with a preparation of silicate of soda and carborundum sand. This should be mixed as follows:

Carborundum fire sand, 65 per cent.
Ground fire clay, 20 per cent.
Silicate Soda, 52 per cent. Baumé 15 per cent.

If this solution shows any tendency to burn off the back wall, leaving the brick exposed in spots, or even taking part of the brick off with it, your burner adjustment or your oil pressure is wrong. If Bureau burners are installed you may be able to draw them out slightly, but in either case you must reduce your oil pressure so that the maximum combustion will

come in the center of your furnace. This can usually be judged by the soot deposit on the tubes, which is thickest near the center of the combustion area. Tubes should, of course, be blown as soon as the boilers are cool, and before putting on the smoke pipe covers, if possible.

In most of our small-tube boiler installations, about forty minutes in warm weather and one hour in cold weather is sufficient time for raising steam without danger to the boiler, but, of course, it can be raised in ten minutes in an emergency.

Each and every installation has its own peculiarities, and the engineer officer will be well repaid who puts time and study into the proper operation of this most vital part of his plant, and the fireroom crew of a modern oil-burning ship requires the same careful drilling and attention that the gun crews get. It is a great advantage to keep the watches together, and to stimulate, if possible, by competition, the interest to excel in economical operation, both underway and at anchor.

NOTES ON PUMPS.

By Lieutenant S. M. Robinson, U. S. N., Member.

The following notes have been compiled from information obtained from officers in the service:

No attempt has been made to describe the various pumps in the service, as it is the intention to use these notes in connection with the actual operation of pumps on board ship where drawings and descriptions of the pumps will be available.

Practically all of the ships in the Navy are fitted with main air pumps of the bucket type, main circulating pumps of the centrifugal type driven by a vertical, reciprocating engine, and service pumps of the plunger type. The *Salem* has a Blake dry-air pump driven by a vertical reciprocating engine and a Worthington hot-well pump driven by a turbine. The *North Dakota* has a Weir monotype wet-air pump; originally she also had a Weir rotative dry-air pump, but that has been removed as it did not prove satisfactory. The *Jupiter* has an Alberger dry-air pump driven by a vertical, reciprocating engine and a turbine-driven Alberger hot-well pump. The *Neptune* has a centrifugal circulating pump and a Westinghouse-Leblanc air pump on one shaft driven by a turbine, and a turbine-driven hot-well pump. The *Wyoming* has a Weir single-acting, dual air pump with wet and dry cylinders. The *Arkansas* and several of the later destroyers have turbine-driven circulating pumps, and practically all the new ships are being fitted with these. The new battleships are being fitted with turbine-driven feed pumps. The *California* is to be fitted with motor-driven main air, main circulating, dry-vacuum, hot-well and forced-lubrication pumps.

Pumps have been supplied to the Government by many different manufacturers, among whom are Dow, Davidson,

Blake, Worthington, Warren, Snow, Cameron, Deane and many others. There are, however, many general rules of care and operation that apply to all of them equally well, and these will be given first and followed by notes of points that are peculiar to various makes.

STARTING AND OPERATING.

The operation of both bucket and plunger pumps is very simple. First see that the proper suction and discharge valves are open, then open the drains, then open the exhaust, and then admit steam gradually. After the pump makes a few strokes close the drains and bring the pump up to the proper speed slowly. If cushioning valves are fitted, these should be kept closed at slow speeds and opened slightly as it becomes necessary to speed up the pump.

If the pump will not start, close the throttle and look for a closed valve on the discharge line; if none is found closed, then disconnect the valve gear, and, after cracking the throttle, work the valve by hand; if the pump still refuses to start, close the throttle and examine the main steam valve to see if it has over-ridden and stuck; if the main steam valve is all right, then the plunger is probably frozen, particularly if the pump has not been in service for some time.

After the pump has been started it may be found that it won't take a suction; this will be shown by the jerky operation of the pump. In this case, first look for a closed valve in the suction line. Or, if it is a feed pump, it may be that the valves have been leaking from the feed line and the pump is vapor-bound; in this case take a suction from a double bottom for a few strokes to cool the pump or turn a hose on it. Most feed pumps are below the feed tank and have a suction head, but some of the older ones have a suction lift, and these may get air-bound and require priming before they will take a suction. The same thing applies to other pumps having a suction lift; those pumping salt water can usually be readily primed from the sea.

If the pump races without any appreciable effect on the pressure gage, then either the plunger leaks or something is wrong with the valves in the water end. It may be that a valve is broken or merely that the valves do not seat properly. If the pump has been running all right and suddenly loses pressure on one stroke, it is safe to look for a broken valve at once. In some cases, with feed pumps running fast, the valve rims have been pushed off and the guides forced down into the water passages. In the case of air pumps, if the pump works with a jerky motion and the vacuum gage follows the motion of the pump, then something is wrong with the foot valves; if the vacuum drops and the pump races, the bucket valves are probably gone; if the pump holds a reduced vacuum and works somewhat jerkily, the trouble is with the head valves. If an air pump works steadily most of the time but gives an occasional short stroke, it will probably be found that the stuffing box of the auxiliary valve is too loose and the weight of the rocker arm is sufficient to move the valve and reverse the pump.

It should not be attempted to run air pumps too slow as they will probably stop; the reason is that practically all the work is done at the end of the stroke and, in slowing the pump down during its stroke, the throttle may be closed so much that the pump will not have sufficient steam to finish the stroke.

Grunting or groaning in the steam cylinder of a pump is usually due to the steam getting behind the piston rings and forcing them out against the cylinder; or to a scored cylinder; or to cut rings; or to cylinders being out of line. Grunting in the water cylinder is generally due to the packing being too tight. The cure for these troubles will be considered later on.

Pumps taking a suction from the sea (or other pumps with considerable suction head) sometimes pound on the water end; frequently this can be stopped by closing the suction off slightly, or, better still, by putting heavier springs on the suction valves.

If air chambers are not fitted on the suction side of pumps

the water end will usually hammer; a snifting valve on the suction side of the pump will usually relieve this.

Always keep a pump running on full stroke; the methods of insuring this will be taken up later on.

In the case of poorly designed pumps that stop continually an oiler will frequently attempt to use a jacking bar for starting the pump with steam turned on it; this is a dangerous practice and generally results in wrecking the pump sooner or later.

SECURING.

In securing bucket or plunger pumps first shut off the steam, then the exhaust and then open the drains. Close the suction and discharge valves and, after the cylinders are drained, close the drains. Drains should never be left open, as the air will cause cylinders to rust. In securing feed pumps they should be tested for defects in the following manner: Bring the pump up to 30 or 40 pounds above the boiler pressure, then close the discharge and shut steam off the pump; if the pressure holds, then the water end is tight; if the pressure falls, then the water end should be examined for the following defects:

(1) Leaky water piston;
(2) Leaky valves;
(3) Weak relief valve spring;
(4) Leaky joints;
(5) Leaky stuffing box;
(6) Defects in the water chamber which allow water to pass.

CARE AND UPKEEP.

All pumps should be moved daily to keep them from freezing. In order to make sure of carrying this out, a pointer should be fitted on the crosshead and seven marks made on a rod which the pointer will pass. Each of these marks should be labeled with one of the days of the week; it is then always easy to see if the pumps have been moved.

Be sure that the steam and water ends of pumps are kept in line. Pumps secured to a bulkhead are much more apt to get out of alignment than those with an independent base and setting, such as air pumps. Failure to keep proper alignment is one of the greatest sources of trouble with pumps on board ship and is one of the most serious defects. A pump may have been nicely aligned in the shop and then pulled out of line when it was bolted to the bulkhead, or, after it was secured, the ship may have changed her shape sufficiently to warp the bulkhead and cause the same result. If the pump is run in this condition it usually results in scoring the rod and cylinders and breaking followers and bolts. The alignment of pumps should be tested occasionally by removing piston and plunger and running a line through the cylinders; especially should this be done during the first year of commission of a new ship and also if a pump is giving trouble by scoring the rod or cylinder or breaking followers.

Pumps should be made to run with a full length of stroke. This means better steam economy and better all around working of the pump, and it invariably means that something is wrong, either with the design or adjustment of the pump, when it cannot be obtained. A short stroke means incomplete cushioning and causes shoulders to form in cylinders and valve chests with resultant breakage of rings and followers. The proper valve setting must be carefully determined to make a pump take full stroke. In this connection the following thumb-rule for setting valves has been tried and found to be satisfactory: Place pistons in the centers of the cylinders or on half stroke and the valves the same. Then with the top collar all the way down to the tappet, the valve should be open at the top ¼ inch and with the bottom collar all the way up to the tappet, the valve should be open ¼ inch at the bottom; with the collars at equal distances from the tappets the valve should be at the center. Another method of setting valves is to place pistons and valve on the center, as before; then move each collar from the tappet one-half the width of the steam

port. In the case of pumps where the tappet moves the full distance of the stroke, the distance from collar to tappet would be ½ (stroke minus steam-port opening). After the valves have been set trams should be made for the collars so they can be reset without removing the valve-chest covers. It is also advisable to drill holes in the tap screws of the collars and run a wire through them after they have been set; this will prevent inexperienced men from moving the collars, and it should not be necessary to continually keep moving the collars while the pump is running; if this is necessary, something is wrong with the pump and it should be dismantled and examined. Of course, some pumps are not properly designed and, when adjusted to give full stroke at low speeds, will override and pound at full speed. The only remedy for this is to have two separate valve adjustments and two sets of cams so that the valves can be set quickly. If a pump runs on short stroke for any length of time difficulty may be found later on in making the pump take a full stroke; it will usually be found that a shoulder has been worn in the valve seat and this will have to be removed before full stroke can be obtained.

Valve gears should be frequently removed and cleaned with kerosene.

Valves should always be kept scraped true and the packing rings of piston valves kept free. It is highly important that the steam valves should be kept tight; if the control valve leaks, the main steam valve will become steam-bound and the pump will stop. In facing off the pilot-valve seat of a Blake pump care should be taken not to face off so much that the valve will not be able to seat itself. With the new Davidson valve gear it is very important to keep the tapered pin in the small exhaust valve set up hard enough to insure a slight friction on the valve chest, otherwise the pin will get loose and the valve will float back and forth on the small play allowed it, and the pump will run on a short stroke and frequently stop; this pin is intended to take up wear and will have to be adjusted occasionally. With the old Davidson gear of

the cam and pin type, the best results were obtained by case-hardening the cam and fitting the pin with a roller and keeping plenty of spares on hand. Main steam valves are frequently put together without lock nuts; in this case they usually fall to pieces after a short time; lock nuts should be fitted and set screws put in to hold the lock nuts.

When the pins or holes in valve levers wear excessively this will seriously affect the valve motion; if this wear occurs rapidly it is best to bush the holes with a tool-steel bushing.

Examine the valves in the water end frequently. When overhauling them, be sure that the springs are in good condition and set up to the proper tension and secured by split pins. See that the valves are true; try them for this with a straight-edge; if this shows the valves to be dished or warped do not turn them, but put in new valves if they are made of metal, as metal valves are apt to break when they have been dished; in the case of rubber valves, their life can sometimes be prolonged by trimming and turning them. Keep the valves clean; a light mineral oil makes a good cleaner and a lye or soda solution is good for removing caked or gummed oil from valves. Be sure that the valves are set to have a lift which will give a circumferential opening equal to the clear opening through the seat; this will never be more than ¼ of the diameter of the opening. In renewing valves the new ones may be of slightly different thickness from the old, thus giving the valve too much or too little lift; too little will usually cause the valves to squeal. In Blake pumps of the valve-pot water-cylinder type, it is very easy to make a mistake in assembling the valves. If the nut on the discharge-valve spring guard is set up before the valve-plate rod nut, then the collar on the valve-plate rod will not bear on the suction-valve spring guard, and the suction-valve plate will be free, and the pump will simply churn water back and forth in the suction chambers. Be sure that the valves and seats are smooth so as to give a perfect bearing surface. Some pumps have the valve seats secured only by making them with a driv-

ing taper fit, and these occasionally work loose; the remedy is to peen over the edge of the metal slightly. Sometimes the valves are placed with several in one plate and the joint between valve plate and pump cylinder is made with a ground joint; it is very difficult to make this joint tight, and if it is not, water will leak under the joint and score it; this can be reseated in a ship's valve reseating machine; a copper gasket for this joint seems to give better results. In pumps that have the valve seats screwed into the pump diaphragm they should always be screwed in with white lead, otherwise it will be almost impossible to get them out. In Davidson pumps the discharge valve seats in the water end are sometimes secured to the pump diaphragm by shoulders on the valve stems which are screwed into the suction seats and these seats have small flanges under which gaskets are fitted. The manufacturer supplies a rubber gasket, but this will very shortly be squeezed out and the seats will leak and hammer. Hard sheet packing will give better satisfaction than rubber, and lead better still if the water to be handled is cold. If sufficient time can be spared for it to set, red lead makes a satisfactory joint. In some cases this flange has given so much trouble that it has been necessary to fit new seats with a ground joint. In air pumps the stems of the valves are sometimes too loose a fit in the seat; this results in allowing the stems to work, and this breaks the wire securing the stems together and then all of the stems back out. This can be remedied by refitting the stems. The most trouble with the water-end valves usually is in the main feed pumps; frequently the stems of these valves will break so often as to become a serious matter. A good remedy for this has been found, and consists in cutting the stem off the valve and turning a groove in the top of the valve for the spring to set in; the stem is pinned to the guard at sufficient height to allow proper opening of the valve, and it then acts merely as a guide for the spring and a limiting device for the valve. It will be necessary to fit a new stem, as the old one will not be large enough to make a close fit in the guard.

In some of the older vessels the air pumps have soft-rubber valves; these get soft and stick in a short time and also collect oil rapidly; they should be overhauled and cleaned after every run and renewed frequently. A strong soda solution is best for cleaning these valves.

The rubber valves in use at the present time generally wear the hole in them too large before they wear out or get soft; by bushing this hole with lignum vitae (boiled in linseed oil) the life of the valve can be very materially increased.

Sometimes the rubber valves in auxiliary air and circulating pumps and fire and bilge pumps disintegrate rapidly, and it has been found that thin bronze plates, similar to those used in air pumps, will give good results in such cases. These valves are usually fairly noisy, however.

Some of the cast valves are quite heavy and give trouble due to heavy hammering; the substitution of light bronze discs will usually relieve this trouble also.

Water plungers should be kept tightly packed but not tight enough to make them groan; in putting in flax or Tuck's packing it is best to soak it before putting it in, but if there is not sufficient time for this, the packing must be fitted loose enough to allow for expansion. Water cylinders should be examined occasionally to see if they are scored and to see if the packing or rings are in good condition. Pump rods are sometimes screwed into the plunger and secured by a lock nut with a flat plate secured on top of the plunger to prevent the nut from backing off. This will still allow the nut and plunger to turn together and the plunger may back off and carry away. This can be prevented by putting a set screw in the nut. A copper washer under the nut will help secure the nut from turning and getting loose on the rod.

The troubles in the steam cylinder mostly come from the rings. With split rings the steam may get in behind the rings and force them out against the cylinder, causing the pump to groan and cutting the cylinder and rings. The remedy is to fit locking bolts which will allow the rings to open

out only to the diameter of the cylinder; it will be necessary to readjust this ring as wear occurs. In aggravated cases relief has been obtained by turning a groove about 1/32 inch deep and 3/8 inch wide about the middle of the ring and drilling 1/8-inch holes right through the ring; these holes will relieve the pressure of the steam. When split rings cannot be made to work, resort is usually had to a solid ring, but this is not very satisfactory as this will leak as soon as it wears a little. Diagonally-cut split rings seem to give better satisfaction than over-lapping rings; the latter generally break at the corner after being in service a short time, although this trouble can be reduced by rounding the corners of the shoulders. Another source of trouble is the breaking of followers and bolts; this may be due to bad alignment, but it is also frequently due to a weak follower; where this trouble is general in any one set of pumps on board ship, it is pretty safe to assume that the followers are too weak and a new follower should be tried on one of the pumps to see if it stops the trouble. This condition can sometimes be improved by fitting the piston with through bolts. Pistons frequently work loose on the rods; this is generally due to poor shop work; the rod will be so fitted that the shoulder brings up against the piston without giving a proper bearing surface for the taper part of the rod; if the rod is properly fitted the trouble will usually stop. If a jamb nut is not fitted, that should be tried also.

Great care should be exercised in setting up on glands; if the two sides are set up unequally the gland becomes tilted and scores the rod and sometimes the gland itself is broken. If a gland is so located that one of the studs is not readily accessible it is best to fit a gland nut so that it will be impossible to tilt it. In the case of outside packed pumps, where the gland is so important, it has been found best to fit a piece of pipe on each stud after getting the gland properly adjusted, in order to prevent inexperienced men from changing it. If a gland is set up too hard it will score the rod; if a gland continues to leak after it has been given a few turns on the nuts

the best thing to do is to let it alone till the pump can be shut down, and then renew the packing. When excessive corrosion takes place on the salt-water end of the piston rods, this can sometimes be reduced by fitting a packing ring in the stuffing box and using a grease cup on it. The use of monel metal for the rods also eliminates this trouble.

Trouble has occasionally been experienced with the relief line from the feed pumps; if this is piped back to the suction line close to the pump, in some cases when the relief valve lifts, it will blow the water away from the suction and cause the pump to become vapor-bound; this trouble can be stopped by taking the relief line back to the feed tank. In the case of an air or vapor-bound pump it has been found that a pet cock for getting rid of the air will be more effective if placed at the top of the air chamber on the discharge side of the pump rather than on the valve-chest bonnet; this is, of course, only true when the air chamber is near the pump. Air chambers should be fitted on both suction and discharge sides of pumps and pet cocks should be placed on both at the highest point to admit air to take the place of that which has been absorbed by the water. Air chambers should be located as nearly as possible with the opening opposite to the direction of flow of the water.

When feed lines are fitted with grease extractors a pipe connection should be fitted on the discharge side of the grease extractor and led to a gage at the feed pump. This will immediately show if there is anything wrong with the grease extractor.

Feed-pump regulators are sometimes connected direct to the water end of the pump, and usually this causes the regulating diaphragm to cut; the trouble can be remedied by piping from the discharge side of the feed heater to the regulator and fitting an air chamber on the regulator.

LUBRICATION.

A very small amount of cylinder oil should be used on the steam ends of the rods and all outside moving parts should be

lubricated with mineral oil. No oil should be used in steam or water cylinders or valve chest.

VALVES.

The present specifications for reciprocating pumps for the Navy calls for metallic valves of either manganese or phosphor bronze for the feed pumps and air pumps and hard or medium-hard rubber valves for all other pumps. The valves for the air pumps are usually rolled and those for the feed pumps cast. The rubber valves are made of vulcanized rubber containing not less than 30 per cent. Para rubber for the hard and 35 per cent. for the medium-hard valves.

PACKING.

The present pump specifications call for metallic packing for the piston rod and valve stem packing on the steam ends of all pumps.

For the water end either flax or Tuck's packing or a combination of alternate layers of the two seems best for the rod and also for the plungers of outside-packed pumps. For water plungers flax or Tuck's seems to be best for all pumps except feed pumps, where white metal rings are generally used. There is a great divergence of opinion as to the best packing for feed-pump plungers, many officers liking Tuck's better than the white metal and others preferring the white metal. The latter seems to be gaining in popularity, however. These rings can be carried in the form of cast cylinders and a ring cut off when desired. If solid rings are used instead of split ones, their tightness can be improved by cutting water grooves about ¼ inch apart, ⅛ inch wide and 1/16 inch deep; these rings should be fitted tight enough so that they will just drop through slowly of their own weight. If split rings are used they should be fitted with springs to hold them out and with a lock piece to prevent too great expansion. A satisfactory mixture for white metal consists of two parts tin and one part lead. These rings as fitted are sometimes very

narrow, and this usually causes them to cut the cylinder; if wider rings are fitted the trouble generally stops.

For steam pistons, diagonally-split cast-iron rings with locking piece to limit the expansion seem to give the best satisfaction. Methods of preventing steam from getting behind these have already been considered.

Piston valves are generally fitted with rings similarly to the steam piston, but these have sometimes given trouble, which has been eliminated by carefully fitting a solid piston valve; this will not remain satisfactory if the valve wears much, however.

For all cold-water pumps a sheet packing of rubber with cloth insertion is suitable for the water end. For the steam ends of all pumps and water ends of hot-water pumps compressed-fiber packing is used. When trouble is experienced with gaskets blowing out the trouble can usually be eliminated by turning a shoulder on the head and cylinder so as to make a male and female joint; this shoulder will prevent the gasket from blowing out. On the water cylinder the head with the stuffing box gives the most trouble by blowing gaskets, on account of an alternate increase and decrease of pressure on the gasket due to the motion of the rod; this will be aggravated if the stuffing box is unduly tight. A very reliable joint for this end can be made by fitting the shoulder as above described and then using a copper gasket cut out in one piece and put on by disconnecting the rod at the crosshead.

CIRCULATING PUMPS.

Circulating-pump engines are usually compound, direct-acting engines; the new battleships have forced lubrication for these engines. On many of the smaller ships the engines are single-cylinder and these are usually fitted with a flywheel; these sometimes give trouble by vibration; this can be remedied by putting a balance weight on the flywheel. It is usual to start a circulating-pump engine from a certain position and a jacking bar is provided for jacking them into position; oc-

casionally some one forgets to remove this bar before starting or attempts to jack the engine over with steam turned on, and this generally results in a serious accident.

Before starting the lubrication should be turned on, the engine jacked into position, the drains opened, the exhaust opened and then steam turned on slowly. Sometimes it will be necessary to give a quick opening to the throttle to get the engine started, but care must be exercised if this is done; make sure that the cylinders are well drained and warmed up. After the engine starts, open the overboard discharge and the main injection.

The speed of a circulating pump should be regulated according to the amount of circulating water needed. As a rule the overboard discharge should be between 10 degrees and 20 degrees F. below the temperature corresponding to the vacuum.

When securing, close the throttle, then the exhaust and then open the cylinder drains and shut off the lubrication. Close the overboard discharge and main injection valves. After the cylinders are drained close the drains; if the steam or exhaust valves are leaking it may be necessary to keep the drains open.

These engines run at a high speed and must be kept carefully adjusted. The cylinders should be opened frequently and wiped out with kerosene. The bearings should be adjusted at the first sign of a knock. After making adjustments the engine should always be run for a short time to see if there is any tendency for the bearings to heat.

These engines should be jacked over daily.

The principal troubles in the water end are due to corrosion of the pump shaft and the runner getting loose on the shaft. The keys on the runners should be secured by set screws. Zincs fitted in the pump chamber will reduce the corrosion.

The lignum vitae bushings for the bearings sometimes wear rapidly, and this puts the pump out of alignment; this should be checked frequently and a new bushing fitted when neces-

sary. This bushing should be well soaked before fitting it, otherwise it will bind when it swells and the shaft will squeal. Forty-eight hours is about the minimum time that can be allowed for soaking lignum vitae.

The pump casing should be removed at least once a year and the pump examined; dirt and marine growth will collect in the pump chamber and must be cleaned out. Every time the ship docks, the pump chamber, suction piping and as much of the discharge piping as possible should be cleaned as this growth very materially reduces the efficiency of the pump.

Sometimes a circulating pump will have an unbalanced end-thrust so that the runner end clearance will not be equally divided and the crank web will rub against the bearing on one side. The best way to overcome this difficulty is to fit a thrust bearing on the end of the shaft. A simple device for this consists of a small iron bracket bolted to the framing and carrying a brass plate which bears against the end of the shaft. The brass plate is held against the end of the shaft by a set screw in the bracket and is prevented from revolving by a stop. Lubrication can be supplied through an oil hole to the center of the brass plate and distributed over its surface by a groove cut across a diameter of the shaft end.

Trouble has been caused by the zinc plates in the condenser dropping down and getting into the pump casing, causing it to rattle badly and sometimes breaking a pump runner. If brass washers are fitted on the studs holding the zincs or through bolts with washers fitted instead of the studs, this will usually eliminate the trouble; also the zincs should be renewed frequently, and this will reduce the probability of such an accident.

SPECIAL TYPES OF PUMPS.

U. S. S. Wyoming—Main Air Pumps.

These are Weir dual air pumps. The main point requiring special attention is a spring-loaded valve on the discharge pipe from dry to wet cylinder. This valve is adjusted to maintain

a vacuum of about 20 inches in dry cylinder when the condenser has 28 inches vacuum, and this difference of 8 inches in vacuum is sufficient to overcome the friction in the cooler and to pass the water into the suction. This valve and spring must be kept in good condition to obtain satisfactory results.

U. S. S. Jupiter—Dry Vacuum Pumps.

These are Alberger, reciprocating-engine-driven pumps. To start up, first see the circulating water turned on; next see that the exhaust valve is open; next see that the atmospheric valve on the suction to the air cylinder is open and that the valve to the condenser is closed; next jack the engine over to starting position; next open cylinder drains; next turn on steam; after the engine has made a few revolutions, close the valve to the atmosphere and then slowly open the suction from the condenser; close cylinder drains. These pumps have an oil-relay governor operated by the vacuum.

The upkeep of the steam end of these pumps is the same as that of any reciprocating engine; they are slow speed and give very little trouble.

The discharge valves on the air cylinder should be frequently cleaned with kerosene as they become covered with a deposit of burned oil.

The air cylinder and piston should be examined and wiped out with kerosene once a month.

Hot-Well Pumps.

These are turbine-driven volute pumps made by Alberger.

These pumps should not be started up until there is sufficient water in the condenser to fill the suction pipes; a gage glass has been fitted on the condenser to show height of water in it. Before starting up make sure that the equalizing pipe from the top of the condenser to the suction side of the pump is open, otherwise the pump will probably be air-bound; also make sure that the ¼-inch water-seal pipe leading from the

discharge side of the pump to the lantern ring in the stuffing box on the water end is open, as otherwise this gland will heat very rapidly and score the shaft, and also the pump will suck in air; there is a pet cock on this pipe which will always show whether or not the pump is throwing water; when running, a little water should trickle out of the stuffing box, and if it does not, the packing is too tight and the gland should be eased up. After these valves are examined see that the oil reservoir is about half full, open the exhaust, see that the suction and discharge valves are closed and turn steam on slowly till throttle is open (the turbine is governor-controlled); then open the suction valve and then open the discharge valve; if this procedure is followed no trouble will be experienced in getting the pumps quickly free of air and to pumping water.

The packing in the water-end stuffing box should be renewed about once in two months to make sure that it does not get hard and score the shaft.

The thrust bearing should be examined about once a month.

U. S. S. Salem.—Dry Vacuum Pump.

This is of similar type to that on the *Jupiter* but was supplied by the Blake Company. The operation of the two pumps is similar, but the dry-air pump connection to the *Salem* is low enough so that occasionally the water level in the condenser gets up to it; this will be shown by a change in the running noise of the pump and by the discharge of water from the drain valve on the bottom of the valve chest (this valve should always be kept open); when this happens, the suction valve should be closed at once, as water in a rotative dry vacuum pump will probably wreck the air end. Then open the drain valves on the air cylinder and it will free itself of water. Then, as soon as the water in the condenser has been removed, the suction valve can be opened slowly and the drain valves closed.

The springs of the air cut-off valves should be examined and renewed frequently. These valves as originally installed

gave trouble due to flapping on the valve face, but this was remedied by putting in guide rods for them. These valves can be adjusted while the pump is running. The point of best running should be determined by trial and marked on the adjustment-gage strip.

Hot-Well Pumps.

These are Curtis turbine-driven pumps of Worthington manufacture.

These are of a similar type of pump to those on the *Jupiter* and the operation is similar to those pumps. These pumps are governor-controlled, but are also provided with hand-operated nozzles which increase the turbine efficiency at the low speeds.

The greatest source of danger is in the water seal; if the pipe gets clogged it will have to be disconnected and blown out; if the gland is not properly adjusted the seal water may be cut off from the seal; this can be prevented by putting a set screw under the edge of the gland and screwed into the main casting.

Care should be taken to keep the roller bearings well oiled and the collars adjusted to give the turbine rotor the proper clearance; this can be measured when the turbine is stopped by removing a cover plate and using a graduated steel wedge.

North Dakota.—Main Air Pumps.

These are Weir monotype. The care and operation of them is practically the same as for other bucket air pumps. They only take steam on the up stroke. When first installed they hammered somewhat at the end of the stroke, due to the fact that the steam pressure on opposite sides of the piston was equalized before exhaust took place allowing the vacuum to pull the plunger down with a slap. This was remedied by fitting a pipe at the top of the air cylinder with both ends of the pipe leading into the cylinder, one end being above the other enough so that when the bucket reaches the end of its

stroke it passes over the lower end of the pipe and thus cross-connects the two ends of the cylinder and equalizes the pressure.

U. S. S. Neptune.—Main Air and Circulating Pumps.

These are horizontal geared-turbine pumps of the Westinghouse-Leblanc type, the air and circulating pump being run by the same turbine with a 7 : 1 reduction, the turbine running at 3,500 r.p.m. The glands of the circulating pump are supplied from the discharge side of the pump and must be kept clear. The air pump has a reservoir for fresh water in its base and a cooling coil for this water which is cooled by the main injection. A valve in the fresh-water line regulates the amount of water to the pump. The air-pump glands are supplied with fresh water from the main feed line and these glands must be kept open and supplied as is the case with all pumps of this type.

Hot-Well Pumps.

These are direct turbine-driven pumps running at about 2,500 r.p.m. The operation and care of this pump is similar to that of the hot-well pumps on the *Jupiter* and *Salem*.

INTERESTING ACCIDENTS AND REPAIRS ON BOARD SHIP.

The main circulating-pump casing of the *Iowa* leaked; it was discovered that the metal where the chaplet rods had been in the mold was spongy; this was tapped out and plugged.

The main circulating pump of the *North Dakota* rattled badly and for a long time no reason could be found for it. The casing of the pump had a very sharp edge at the junction of the discharge pipe and the end of the spiral; this was smoothed up and the rattling ceased.

The water pistons of the main feed pumps of the *Vermont* originally consisted of a solid brass ring held in place by a follower plate; this piston was unsatisfactory and kept the cylin-

der scored all the time and was not tight; the brass ring was replaced by a sectional-cast white-metal ring with springs behind it, and this proved satisfactory. These rings were 6 inches wide.

On the same ship, while cruising at a long distance from a navy yard, two of the steam chests of the main feed pumps broke off short at the neck of the flange and on the end where the bell crank for working the valve gear was attached. In breaking, half of the flange remained attached to the steam chest and the other half to the bonnet. Repairs were made by the ship's force. The broken pieces of the flanges were put in place and an iron band fitted and shrunk around the flange to hold it in place; then long stud bolts were fitted from end to end of the flanges to hold the broken parts of the steam chest together. A sheet-iron casing was then fitted around the flange to about half the length of the steam chest, leaving a space of about two inches between the external side of the steam chest and the sheet-iron casing. This space was utilized for making a rust joint made of a mixture of cast-iron borings, sulphur, salammoniac and water. After setting for 24 hours the joint was solid and held when placed in service. This method of repair has been used on other ships in a number of cases for both valve chests and cylinders. When possible to do so, it is best to caulk copper wire into the crack on the inside of the cylinder. A good mixture for this job consists of 1 pound iron borings, 2 ounces of salammoniac powder, 1 ounce of flowers of sulphur; mix the whole dry; use this mixture with 20 pounds of iron borings, mixing with water to the consistency of mortar.

On the *San Diego* one of the pumps always had a heavy thump on the bottom stroke. The trouble was finally located in the liner; it did not fit at the bottom of the cylinder and there was a ¼-inch hole through the cylinder wall connecting to the suction port. The hole was plugged and the thumping stopped.

On the *Resolute* the spiders that held the packing in the

water plunger out against the walls of the cylinder carried away. The ship had no spares and no machine shop. Pieces of wood were cut to fill in the space between the piston head and packing; when the wood swelled it held the Tuck's packing out against the cylinder and gave satisfactory working.

Some of the older ships, fitted originally with wick feed lubrication, have been changed to forced feed. The result of the change in some cases was to make it impossible for the throttle man to see whether or not his auxiliaries were running. To overcome this tell-tale lights were fitted up by putting a make-and-break connection on the beam of the air pump and on the indicator connection of the circulating pump.

On the *San Diego* trouble was experienced with the auxiliary air and circulating pump; it stopped frequently. It was found that in putting a stud into the valve chest the button had been punched out of the hole; when steam was turned on this button would blow onto the port. It took some time to discover this, as the button would fall away out of sight as soon as steam was shut off.

On the *San Diego* the hand reversing gear of the main engines did not work satisfactorily, and finally the hand pump was changed as follows: The oil reservoir was removed, the packing of the oil piston was changed to Tuck's, and a connection brought from the discharge side of the main feed pump to the hand-pump plunger cylinder. An exhaust connection was made from the main feed tank to the nozzle on the pump formerly used as a reservoir connection. Thus the oil pressure produced by the hand pump was replaced by the feed-line pressure. This gave good satisfaction.

In the water end of the *Connecticut's* feed pumps the valve stems broke frequently. They were screwed into the suction seats and had a collar for securing the discharge seats. The pump diaphragm between suction and discharge valves was weak and would pant, bringing a heavy strain on the stems and breaking them. Screw stays were fitted between the seats and this stopped the trouble.

On the same pumps the flanges on the cylinder bonnets, chests, and also on the distance pieces between the water legs were not wide enough and gaskets would blow out continually. Heavier bonnets were made and as many stud bolts as possible fitted and this relieved the trouble.

These pumps were outside-packed and the stuffing boxes were too shallow; this resulted in broken gland studs due to attempts to set up too hard on them and also in scored plungers due to too much pressure on the packing. The trouble could not be eliminated, but was reduced by using a special packing of vulcanized rubber made in rings to fit the stuffing boxes exactly.

NOTES.

RECENT PROGRESS IN MILITARY AERONAUTICS.*

By Lieut.-Col. Samuel Reber,

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The first decade of the twentieth century has added to the mechanical engines of war two types of apparatus, one heavier than and the other lighter than air. While the rigors of warfare will be apparently increased by their use, they will prove potent factors in the disappearance of war from the forum of the world as the ultimate means of arbitrament for international controversies. The application of mechanical flight to warfare is but an added proof of the maxim of the German strategist who said, "War is the only science that lays under tribute all other sciences." Any progress in mechanical arts that tends to shorten the duration of a war, even by increasing the destructive potentiality of its implements, is a real gain to humanity, and the discoverer of a process or the inventor of an engine which increases its horrors and destruction is a real benefactor to mankind, since the increased loss of life and property will certainly cause civilized nations to hesitate and seek other less drastic methods for the settlement of international differences before final appeal is made to the arbitrament of arms. The art of mechanical flight has extended the domain of warfare into three dimensional spaces, has created new means of observation, communication and attack, whose potentialities are now but partially realized by the great military powers, whose existence has brought into being a new arm of the service, and whose utilization has produced great changes in the military and naval policies of the great powers.

As sea power has been one of the dominating factors controlling the policies of great powers in the past, air power will be so in the future. The moral effect of the presence or absence of an efficient air force will be a deciding factor in hostile operations, and the commander who is without an efficient air fleet will be at a great disadvantage from the outset of operations, and will have the celerity and certainty of his movements greatly hampered by the feeling that his plans and disposition of his forces are known to his opponent through aërial reconnaissance. To achieve success without an aërial force, he must greatly outnumber his opponent. Information of the location and movements of the enemy are absolutely essential to any commander. Such information is usually obtained by the active use of cavalry in contact with the enemy or by secret service within his lines. An efficient air force will not only supplement but anticipate the reconnaissance work of the cavalry, and can penetrate miles beyond the line of contact, provided it is not opposed by similar and better aircraft of the enemy.

The two best examples as to what might have happened had an army been supplied with a fourth arm were in the recent Russian-Japanese

*Presented at a joint meeting of the Mechanical and Engineering Section and the Aëro Club of Pennsylvania, held March 26, 1914.

war. Had the Russian commander had an aërial reconnaissance force at his disposal it is more than probable that the results of the two decisive battles of Liao Yan and Mukden would have been entirely different. Had Kuropatkin checked the Japanese advance on the Tai-tse-ho in September, 1904, the outcome of the war would have been much more favorable to Russia. On August 25, 1904, the Japanese attacked the Russian position in front of Liao Yan, and on the 27th the Russians, after an obstinate defence, retired to a second position, which was attacked without results until the 31st, when Oyama decided to send the first army under Kuroki to turn the left wing of the Russian position. On the 31st of August and 1st of September Kuroki's army moved to the northeast and separated itself completely from the main position by a distance of ten miles. This movement was absolutely unknown to the Russian commander, and, by its attack on the 2d of September, so relieved the situation along the main front that the Japanese advance, which had been completely arrested by the Russians, was renewed and Kuropatkin compelled to retire along the railroad to prevent his army being cut off from its line of communication. Kuropatkin's information had led him to believe that he was outnumbered at the beginning of the engagement by the Japanese, while, in fact, he had a decided superiority in men and guns. It must be evident that a few hours of aërial reconnaissance would have disclosed the real state of affairs and indicated the isolated position of Kuroki on September 1, and given Kuropatkin a splendid opportunity to capture or defeat with ease the entire first army. Again, in front of Mukden, Kuropatkin was unaware of the arrival of the third army, and even of the existence of the fifth army, nor did he have any information as to the extensive flanking movement of the third army from the 21st of February to the 1st of March, which movement ultimately compelled his withdrawal from his fortified lines. One single speed scout could have started from Kuropatkin's headquarters, circled the country over which the flanking movement was made, and returned and reported the movement within an hour and a half.

The functions of aircraft in war are fourfold: reconnaissance, both strategical and tactical; the prevention of similar reconnaissance on the part of the enemy; for communicating purposes, and for destructive action against the enemy.

The strategical reconnaissance by aircraft will cover all operations for the acquirement of information as to the location and movements of the enemy, and also of the terrain. When the reconnaissance becomes tactical in nature, as for example during an engagement, it will ascertain the location and disposition of the opposing troops, especially the reserves, and artillery positions, and inform the artillery commander of the effect of the fire of his guns. Whether reconnaissance will be made by dirigibles or aëroplanes will depend upon the available aëronautical material at the disposition of the commander general, the weather conditions and the special object of the particular reconnaissance. For long-distance work, and especially for over-sea operations, the dirigible possesses certain desirable features, especially as part of the work is to be done at night. For the prevention of reconnaissance on the part of the enemy there will be required types of fighting machines which can carry light automatic guns and possess a greater climbing ability than those of the opposing force.

For communication purposes the aëroplane is especially suitable. The conditions of modern warfare—the large extent of territory over which forces operate, and the length of the line of contact in a modern battle—render it extremely essential for the commanding general to have the swiftest means of communication, which is best furnished by aëroplanes.

The use of the aëroplane and dirigible for destructive operations offers a large field in the attack by explosives from overhead of permanent

works, depots of supplies, dockyards, railroads, cities, and troops in position, but in the latter case the effect will be more moral than physical. A squadron of dirigibles or aëroplanes can compel a blockading fleet to remain a safe distance away from the shore, keep it constantly in motion, and cause greater coal consumption—a dominating factor in naval operations.

The possibilities of destructive aërial attack are well given in the following extract from a recent paper on aërial warfare by an officer of the French Army, Lieutenant Sensever:

"We shall see later on the possible utilization of the German aërial fleet for the destruction of works of art, arsenals, stations, or the direct struggle against enemies terrestrial or aërial. For the destruction of a city no problem of target practice arises; all shots count; the target is vast; no projectile is lost; no problem of defence is necessary for the aërostat, because in the night time it is invisible; no cannon can reach it; no aëroplane can pursue it; it is heard, but cannot be seen. Let us imagine on the evening of the declaration of war the ten Zeppelins, now in service, set forth with lights extinguished at an elevation of 1,000 meters towards Paris., The compass, the luminous signals of the stations and railroads, the lights of the cities are guides which indicate the direction to an aërostat more clearly, in fact, by night than by day. In the silence of the country, in the nocturnal quietude after the sounds of the city, the distant roar of the motors will vanish, leaving behind it incertitude. What is there to do? The Zeppelins steer towards Paris. In four hours, if there be no wind, they will cover the distance which separates the Rhine from the French capital. Now they are above Pâris, whose innumerable lights mark out the quarters, the squares, the avenues; then it is very simple. When each one arrives at its appointed spot it accomplishes its task of destruction and death: to one, the stations where our soldiers will embark tomorrow; to another, the Ministry of War, the Elysée; to others, the forts or barracks at Versailles or Vincennes; to others, the Opera, the Louvre, Notre Dame, the Arc de Triomphe—all our glories and our forces! In a few minutes each Zeppelin has discharged its four or six bombs of 600 pounds each, in all about 50, or about 30,000 pounds of trinitrotoluol, which will level the beautiful quarters, which will kill millions of sleeping people, which will destroy gas mains, sowing conflagration and confusion in the terrorized city. During these few moments the Germans, from their nacelles, will contemplate their work—this spectacle that Nero never dreamed of. Then they will take their journey towards the east. At daybreak they are resting in their hangars along the Rhine, ready to start out again tomorrow."

The above is not an idle speculation, but could happen if the French are unprepared to meet the invasion of their frontiers through the air. England has this same menace threatening her across the North Sea.

Since the beginning of the twentieth century aëronautics have been used in three wars: the Russia-Japanese, the Bulgarian war and the Turko-Italian, and in minor expeditions of the French and Spaniards in Africa. The dirigible had not been sufficiently developed and the aëroplane was not in existence in the Russia-Japanese war, in which the Russians used captive balloons to advantage for observation purposes and for the direction of their artillery fire. The Japanese, though in possession of a balloon train, made practically no use of it during the war, though at Port Arthur there was an excellent opportunity for its use in locating the Russian fleet. Had the Japanese used a captive balloon the fleet could have been easily located and the siege expedited without the terrific loss imposed upon them by the capture of 203 Metre Hill, from which they obtained a view of the harbor and the ships. In the Turko-Italian war in Africa the Italians used both dirigibles and aëroplanes. These dirigibles, P-2 and P-3, only 4,100 meters in size, made in all 91

flights, and P–3, engaged in the battle of Zanzur, discovered and routed a force of Turko-Arabic cavalry by dropping bombs. These dirigibles, assisted by aëroplanes piloted by Italian officers, were of the greatest value for reconnaissance purposes, and, though frequently exposed to hostile fire, escaped without material damage.

In the Balkan war it may be said that practically no effective use was made of aëroplanes by any of the belligerents, as they were entirely unprepared for this class of service. In the first war the Turks had two in front of Kirk-Kilissé which were broken down and did not fly. They had some others in the rear which were not used until after the Bulgars reached Chataldja. The Bulgars had no trained aviators, and were compelled to hire foreigners, who rendered them practically no service. The advantage to the Bulgars of a single aëroplane is illustrated by that period of loss of contact, October 24 to 27, 1912, when the Bulgarian cavalry broke down and they had no knowledge of the Turkish position.

At Adrianople the Turks had no aëroplanes. The Bulgars used a number which were mostly manned by foreigners. General Ivanoff summed up their use as follows:

"The best that can be said for the aëroplane is that the flights showed what great service they are capable of if well manned."

The Servians, while having four aëroplanes, did not put them in service. At Chataldja a number of flights were made for the Turks by foreign aviators, and the Greeks made several reconnaissance flights in their operations.

The lesson from the Bulgarian war is that material cannot be obtained and military aviators trained after the opening of hostilities, and that foreign or civilian aviators cannot be depended upon for efficient service. Experience has shown that for military aviation the pilots as well as the observers must be trained soldiers.

The first really successful ascent of a dirigible was made in September of 1884 by the "La France," built by Captains Reynard and Krebs. This balloon was only 51 meters in length, with a cubic capacity of 1,864 meters, and driven by a 9-horsepower electric motor, giving it a speed of 14½ miles per hour. It was not until the beginning of the twentieth century that the French began the development of the present non-rigid type of balloon and added this to their military forces. The Germans, whose attention was drawn to the remarkable progress made by the French since the beginning of the century, began the development, under Count Zeppelin, of the rigid type of balloon. Experience has developed three separate types: the rigid, the semi-rigid, and the non-rigid, each possessing certain advantages and disadvantages for military purposes.

The rigid type is composed of a rigid frame of aluminum or wood in whose interior are a number of gas bags. The frame is covered by a light, permeable material, and the nacelles, carrying the motors, equipage and crew, are attached to the frame. The Zeppelins have a frame of aluminum, and the Schutte-Lanz and Speiss types of wood. The envelope of the semi-rigid is completely flexible, and its form is maintained by the pressure of the inclosed gas, while the weight of the nacelle and the equipment is distributed along the envelope by means of a keel or longitudinal girder, as in the Gross, Lebaudy and the Italian military dirigibles and those of the Forlanini type.

In the non-rigid the nacelle is directly suspended from the envelope, as in the Astra Torres, Parseval, Clement-Bayard, Zodiac and the Siemanns-Schukert.

The rigid type possesses the advantage of greater speed, more inherent stability, and, on account of the construction, the possibility of carrying armament not only in the nacelle but on the upper surface; is less liable to damage from hostile fire, owing to the fact that the gas is not contained in one but several separate bags. The rupture of one does not

cause the loss of the entire volume of gas. The rigid carries a larger crew and can remain longer in the air than the other types. The rigid type possesses the disadvantage of high cost, is more difficult to handle in starting and landing, requires a larger number of trained men in handling it in these operations, must be housed when not in motion, is more liable to damage or destruction in unfavorable weather, and more liable to accident, usually total destruction, in making bad landing, as evidenced by the accidents that have happened to the Zeppelins.

The semi-rigid offers a greater resistance, has less speed than the rigid but greater weight economy for equal volume, and is more easily handled and much less liable to damage in landing. It, as well as the non-rigid, can be anchored in the air when not in use, and is transportable, although the keel renders it less so than the non-rigid. The non-rigid has a slower speed in the air than the semi-rigid, is the easiest to handle, has the greatest economy of weight, and possesses the least liability to damage in landing, and is easily repaired and transported. The tendency is towards two types: first, the large, rigid, armed cruiser type, with a radius of action of 500 miles and an endurance in the air of thirty or forty hours. This type is assigned to permanent stations along the frontiers, with a zone of observation and defence. The second is a small, non-rigid type for over-sea operations. The Germans have developed this type of balloon more rapidly than the French and English, as it seems to be part of the German policy to offset England's supremacy on the seas by a German supremacy of the air. There are under construction in Germany dirigibles of this type up to 30,000 cubic meters capacity, with a lifting power of fifteen tons.

Since what may be called the real beginning of aviation in the year 1909 (Fig. 1) the records indicate a tremendous development in speed, distance, duration and altitude. The relative importance that the various great powers have attributed to aëronautics in war may be inferred from the following (Fig. 2), which shows the amounts of money expended by them during the past five years ended in August last.

Our own country was the first to realize the military importance of the aëroplane, and in 1907 the War Department issued specifications and proposals covering the construction of an aëroplane for the military service. In 1908, in the trials at Fort Myer, Va., due to an unfortunate accident, Lieutenant Selfridge was killed and Mr. Orville Wright injured, and activities were suspended until the following year, when the Wright Brothers demonstrated their machine, which was accepted by the Government. Practically nothing was done in this country until after July, 1910, when the first appropriation of $125,000 was available. A few machines were purchased and a few officers placed under instruction at the aviation school at College Park, Md. In 1911 a small aviation detachment was sent to the Texas border at the Camp of Concentration, and a number of reconnaissances made of the Rio Grande frontier. In 1912 they were used in the maneuvers in Connecticut and also for the observation of artillery fire at Fort Riley, Kan. In 1913 they were again sent to the concentration camp at Texas City, Texas, and the school detachment subsequently moved to the present establishment at San Diego, Cal., where the training has been continued and detachments sent to the Hawaiian and Philippine Islands. While the number of officers on this duty has been small and the equipment extremely limited, the results obtained by them, I think, are on a par with anything that has been achieved in the military establishments abroad.

Aviation has appealed to the *esprit* of the French nation in a stronger way than to that of any other nation. The French have led in the development of military aviation perhaps to offset the German superiority in dirigibles. Military aviation in the French Army was in 1912 made a separate arm of the service, it having been under the Engineer Corps.

It is directly under the War Ministry, and has a general officer at its head. The personnel is composed of officers and men recruited from all arms of the service, and when the new organization is completed the French Army will be in the possession of twenty-seven field detachments, five fortresses, and six coastal detachments, for which 450 machines and twenty-five dirigibles are altogether available. The service is divided into two branches—that of the dirigibles and that of the aëroplanes. The headquarters of the French aëronautical service is in Paris, and the whole of the military aëronautical establishments in France and Northern Africa are contained in three groups, with headquarters, respectively, at Versailles, Rheims and Lyons. The field detachments are assigned, as a rule, to the headquarters of the various army corps. The Central Supply

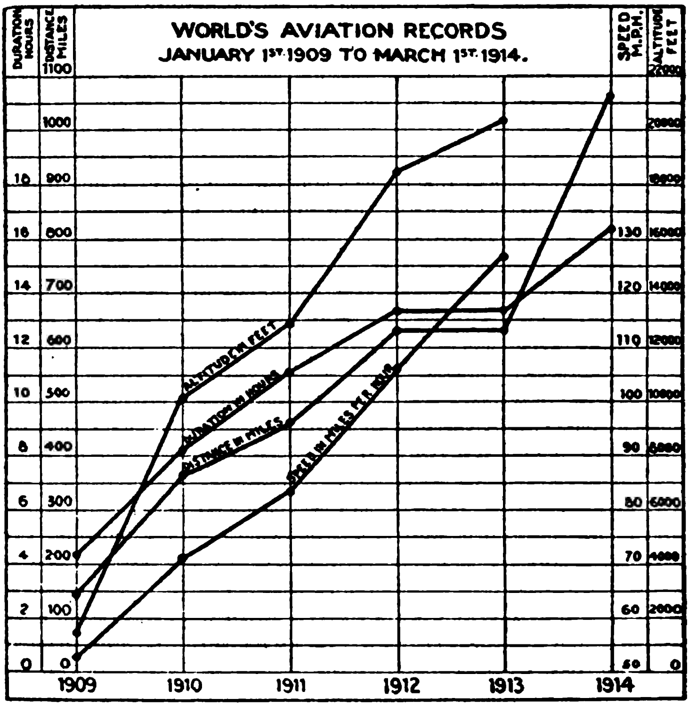


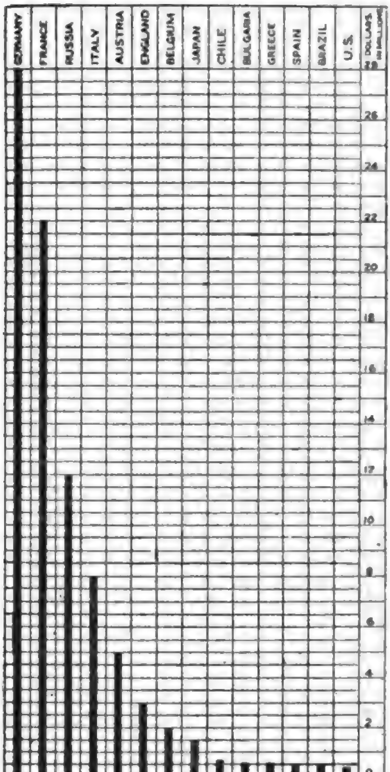
FIG. 1.

Depot and aëronautical laboratory is at Chalais Meuden. The personnel of the flying corps at present is 248 officers and 1,374 enlisted men..

The military aëronautical troops of Germany are under the command of a general of division, and the service is divided into that of dirigible balloons and that of aëroplanes, each under a separate head at Berlin. The total personnel is 82 officers and 1,458 non-commissioned officers and enlisted men. Germany is divided into three grand groups, the first facing towards Russia, the second towards France, Colmar, Metz, Strassburg and the Rhine, the third constituting the entire central hub near Berlin. These troops are divided into four battalions. The first battalion is located near Berlin and at Dresden, the second at Posen, Graudenz and Königsberg,

the third at Cologne, Hanover and Darmstadt, and the fourth at Strassburg, Metz and Friedberg. Bavaria will have a separate battalion. The present aëronautical project calls for an expenditure of twenty millions by the first of January, 1918.

In Great Britain the military aëronautical establishment consists of the Royal Flying Corps, which comprises the central flying school at Upavon, Salisbury Plain, the naval and military wings, the reserve, together with a separate establishment, the Royal Aircraft Factory, and the government aëronautical laboratory. Control of the military policy is under the Air



TOTAL GOVERNMENT EXPENDITURES FOR AERONAUTICS_1908-'13.

FIG. 2.

Committee, which is a sub-committee of the national defence committee. The military wing of the Royal Flying Corps is under the newly-created Department of Military Aëronautics in the War Office, while the naval wing is now managed by the air committee of the Admiralty. So far, no one has been appointed to supreme command of the entire flying corps, as was contemplated in the original scheme, although there is at present some agitation for the establishment of an air department, with a responsible head filling the functions of Minister of Aërial Defence.

There are five aëro squadrons in the military wing, three of them located at South Farnborough, one at Montrose and one at Nethersdon, and eight stations for the naval wing. There are available for service six dirigibles and one hundred and sixty-one aëroplanes. On the first of January of this year all the dirigibles were transferred from the military to the naval wing of the Royal Flying Corps.

Experience has developed three types of aëroplanes for military purposes: The first, the speed scout, for strategical reconnaissance, a one-seater, with speed up to 85 miles per hour and radius of action of 300 miles and a fast climber, about 700 feet per minute; the second for general reconnaissance purposes with the same radius of action, carrying both pilot and observer, and equipped with radiotelegraphy, slower in speed, about 70 miles per hour, and climbing about 500 feet per minute, and in some cases protected by armor; the third, or fighting craft, armored, and carries in addition to the pilot a rapid-fire gun and ammunition, and so arranged as to have a clear field of view and fire in either direction up to 30 degrees from the line of flight, the speed to run from 45 to 65 miles per hour, and the machine to climb about 350 feet per minute.

From the foregoing it is easy to infer what the position of the United States is with respect to air power among the nations of the world. We, who in the beginning started the movement, are now at the tail of the procession. We have no dirigibles, but very few trained men, and fewer machines. The manufacturing industry is moribund from the lack of business, and there is no future for it. We have no aërodynamical laboratories in which to study the problems, and no engineering courses, except one, in which to develop our constructors. The Government has not stimulated any advance in the design of machines or motors by competition for substantial reward. We have no National League, as in France and Germany, to assist the Government by private subscription and by public demand for the development of air power. The interest of our people in aëronautics at large is dead, and has been perhaps so lulled by a sense of false security and the belief that war will not come to such a vast and powerful nation as ours that it will not heed an oft-quoted maxim of the Father of Our Country, "In time of peace prepare for war." In no particular is it more impossible to make up deficiencies after the outbreak of hostilities than in aëronautics. What is to be done? To call to the attention of our people and of our Congress the exact state of affairs today. This can well be achieved through the membership of your institution.—"Journal of the United States Artillery."

THE DIESEL ENGINE IN AMERICA.

By Max Rotter.*

SYNOPSIS.—A résumé of present American practice as regards details of design, from a paper presented at the International Engineering Congress.

"Ignition by compression," so generally and superficially accepted as the fundamental principle underlying Dr. Diesel's invention, was, as a matter of fact, merely a natural corollary, and was not even claimed by him as his original conception, although typical of the engines that bear his name and successfully employed in them alone. Dr. Diesel's great aim was a closer realization of the theoretical efficiency of the Carnot cycle than attainable by steam engines or internal-combustion engines of the ordinary types. And although his aspirations were crowned with only

*Chief Engineer, Busch-Sulzer Bros. Diesel Engine Co., St. Louis, Mo.

partial success, it is scarcely to be doubted that they created a heat engine that has the highest efficiency that will ever be attained with fuel.

Carnot's law, as applying to heat engines, is well understood—the higher the initial temperature and the lower the terminal temperature, or the higher the initial compression pressure and the lower the terminal pressure, the higher will be the theoretical thermal efficiency. Dr. Diesel demonstrated that, although the theoretical and thermal efficiency increased slightly beyond this point, the highest mechanical efficiency was attained with a compression pressure of 30 atmospheres and a compression temperature of 500 degrees C. and, balancing these advantages against one another, that the highest actual efficiency and the greatest useful power development per unit of cylinder volume would be obtained by compressing the air to a pressure of between 30 and 40 atmospheres and a temperature between 500 and 600 degrees C.; and introducing the fuel in such manner as to obtain combustion at constant pressure with a maximum combustion temperature between 1,600 and 1,800 degrees C. On either side of these limits the actual efficiency of the engine diminished. It is evident that the stated compression temperature is " far in excess of that required" solely for the ignition of any carbonaceous substance that would ordinarily be used as fuel; also that, to avoid ignition during the progress of the compression, the fuel must not enter the cylinder until after the full compression has been attained.

In 1894 the first successful experimental engine operating upon these principles was completed by Dr. Diesel. In 1897 Adolphus Busch, of St. Louis, purchased outright the sole rights to all Dr. Diesel's existing and future United States patents and introduced the Diesel engine into this country under the sponsorship of the late Col. E. D. Meier.

The first Diesel engine built in the United States was completed in 1898. Since then these engines have been installed in municipal and commercial central power stations, in the most diversified industrial plants, and as stand-bys for water powers, in capacities up to 1,000 H.P. In Europe stationary Diesel engines have been installed in units up to 4,000 H.P., but the relatively lower prices of fuel in the United States tend to retard a similar development here.

Following in the wake of the Diesel, or high-compression constant-pressure oil engine, has come the so-called semi-Diesel, or low-compression oil engine. The distinguishing characteristics of the two types may be briefly stated as follows:

In the Diesel the entire cylinder volume of pure air is compressed to the maximum pressure of about 35 atmospheres and the corresponding temperature of about 500 degrees C., the fuel is introduced at or about the completion of this compression, and is gasified and ignited by the heat of compression; the combustion taking place without any material increase in pressure, but with a considerable increase in temperature, during 8 to 12 per cent. of the piston stroke, and continuing during a subsequent period of isothermal expansion.

In the semi-Diesel engine the entire cylinder volume of pure air is compressed to about half the compression pressure usual in the Diesel; a small portion of this air being contained in an auxiliary chamber that is in open communication with the interior of the cylinder and that has been heated to a high temperature (by external means prior to starting, or by the heat of the previous combustion while running), the temperature resulting from the mechanical compression being, therefore, considerably higher in this chamber than in the main portion of the cylinder. The fuel is introduced at or about the completion of this compression, either directly and entirely into this auxiliary chamber or through the cylinder and partially into the chamber, and is gasified and ignited by the heat of compression of the air in the chamber, the combustion taking place suddenly and with a greater increase in pressure than in the Diesel and being followed by a more rapid drop.

In engines of both types the maximum ignition pressure is about the same, but the combustion temperature at the commencement of the adiabatic expansion is higher in the Diesel. Figs. 1 and 2 show, respectively, typical indicator diagrams of a Diesel and a semi-Diesel engine.

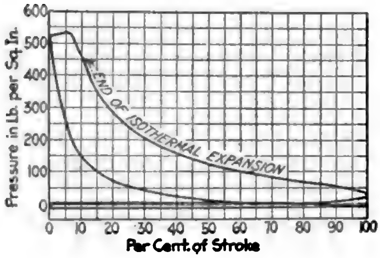


FIG. 1.—DIAGRAM FROM FOUR-STROKE CYCLE DIESEL ENGINE.

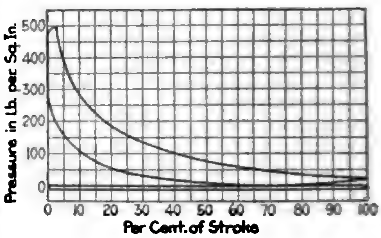


FIG. 2.—DIAGRAM FROM FOUR-STROKE CYCLE SEMI-DIESEL ENGINE.

The two-stroke cycle, as well as the four-stroke cycle of operation, is employed in Diesel engines. The lower efficiency of the two-stroke cycle is due primarily to the power consumed by the scavenging pump, practically none of which is recovered in the working cylinder. This is partly offset by the reduced mechanical losses due to lighter moving parts. The drop of pressure at the release, when the exhaust port of the two-stroke cycle engine is opened, is partially counter-balanced by the "bottom-loop" negative work of the four-stroke cycle.

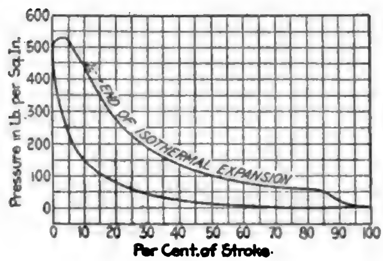


FIG. 3.—DIAGRAM FROM TWO-STROKE CYCLE DIESEL ENGINE.

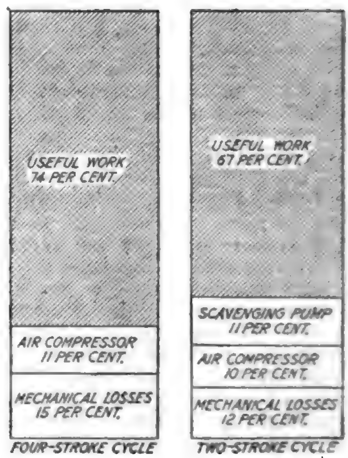


FIG. 4.—HEAT DISTRIBUTION IN TWO- AND FOUR-STROKE CYCLE ENGINES.

Fig. 3 shows a typical diagram of a two-stroke cycle Diesel engine and may be compared with Fig. 1, which is that of a four-stroke cycle engine. Fig. 4 illustrates the proportional subdivision of the indicated power developed by four-stroke cycle and two-stroke cycle Diesel engines.

The curves shown in Fig. 5 compare the fuel consumption of four- and two-stroke cycle Diesel engines built by the same American manufacturer.

The older designs of two-stroke cycle engines, which were provided with cam-operated scavenging and exhaust valves, were run at speeds considerably below those which are usual in the equivalent four-stroke cycle machines, because the camshafts of the former revolve at the same speed as their crankshafts, whereas those of the latter revolve at but half the speed. In later designs this limitation has been practically overcome by the substitution of scavenging and exhaust ports in the cylinder wall, controlled by the piston, in place of the heavy valves. This construction, although familiar in gas-engine practice, was adopted hesitatingly for Diesel engines, on account of their somewhat high exhaust temperatures, the supposed difficulty of constructing suitable cylinders, and the inefficient scavenging resulting from the fact that the scavenging ports had to be uncovered after the exhaust ports were open, and therefore re-covered before the exhaust ports were closed.

In Fig. 6 is illustrated the construction of a modern two-stroke cycle Diesel-engine cylinder provided with a recently patented arrangement of scavenging and exhaust ports. The exhaust ports are controlled by the

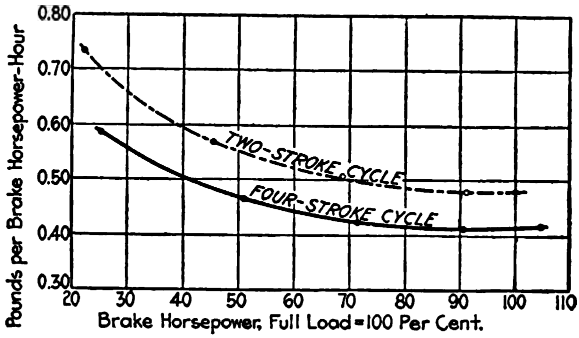


FIG. 5.—CONSUMPTION CURVES FROM SNOW ENGINE.

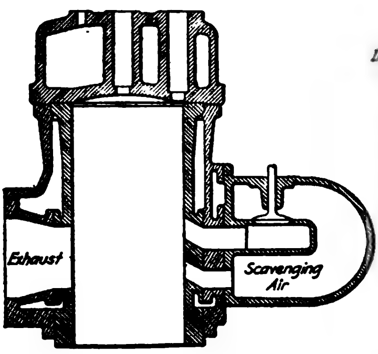


Fig. 6.

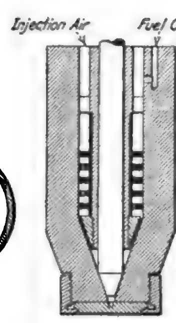


Fig. 7.

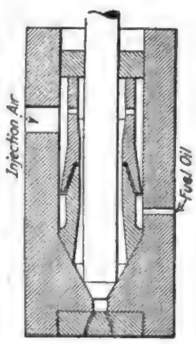


Fig. 8.

FIG. 6.—CYLINDER OF TWO-STROKE CYCLE ENGINE WITH AUXILIARY SCAVENGING PORTS. FIG. 7.—COMMON TYPE OF CLOSED FUEL NOZZLE. FIG. 8.—HESSELMANN FUEL NOZZLE.

piston, as are also the scavenging ports in the lower tier; while the scavenging ports of the upper tier are under the supplemental control of a timed, mechanically-operated valve. The lower tier and the mechanical valve open only after the exhaust ports have been uncovered long enough for the pressure in the cylinder to have fallen below the scavenging air pressure; the mechanical valve remains open after the exhaust ports have been closed. This arrangement not only prevents the dangerous blowing back of hot gases into the scavenging air passages, but also insures more thorough scavenging and the building up of a slight excess pressure in the pure air in the cylinder at the commencement of the compression stroke, thus enhancing its volumetric efficiency and power capacity.

Stationary Diesel engines may be roughly divided into slow-speed and medium-speed machines. High-speed engines of this type are confined to special purposes, such as the propulsion of submarine vessels. Without material variation in piston velocities, the speeds of both classes of stationary engines have been, more or less, selected to conform to the speeds of standard engine-type electric generators. The higher rotative speeds that are becoming general in Europe have, up to the present, not found favor with the American builder, despite the fact that they appear to be entirely successful. The following table compares the speeds of well-known European and American Diesel engines of about 500 H.P.:

	European.	American.
R.p.m.	240	150-200
Piston speed	4.15 m. per sec.	3.8-4.15 m. per sec.
Piston speed	815 ft. per min.	740-815 ft. per min.
Ratio, diam. to stroke	1:1.13	1:1.34-1:1.28

HORIZONTAL VS. VERTICAL ENGINES.

Within these limits there does not appear to be any appreciable difference in efficiency, reliability or wear in engines of equally good design and construction.

In Europe there has been a very general adherence to the vertical arrangement of Diesel engines, and of the few builders of horizontal machines the majority build vertical engines also. In America both styles have their champions, but at the present time the greater number appears to be following the lead of Europe.

There are comparatively few double-acting engines of the Diesel type in operation in Europe and considerably fewer in America. Practically all of these are horizontal machines. It may be reasonably asserted that there is not yet available a sufficiently extensive practical experience with this style to warrant the assurance of success in everyday service; and that, in spite of the many advantages which such an engine would possess, especially in large units, its development still lies well in the future.

Vertical Diesel engines are being built both with "A" frames and with inclosed crank cases. Coincidently with the tendency toward higher speeds, there has developed in Europe an increasing inclination toward the inclosed crank case; and Sulzer Bros., of Switzerland, recently exhibited in Berne a Diesel engine of this construction, of 1,000 B.H.P. rated capacity. The inclosed crank case for Diesel engines was adopted in America long before it received any consideration in Europe, and the original American builders still adhere to this style of frame for engines of small and medium capacities.

Practically all Diesel-engine cylinders are now built up of a liner, or inner barrel; a jacket, or outer cylinder, and a cylinder head. The space between the liner and the jacket forms the water jacket. The purpose of this construction is less that of facilitating the renewal of the inner barrel should it become worn and of permitting its free axial expansion,

than to enable the use of the most suitable iron to withstand the temperatures to which the cylinder is subjected, without the manufacturing risks involved in casting.

The admission and exhaust valves, which are exposed to the high combustion temperatures and to the corrosive action of the fuel, most of which contains sulphur, are now generally made up of a head of hard cast iron, with a stem of forged steel. In this country Diesel engines are not yet being built in sizes that would make the water cooling of the exhaust valves either necessary or convenient, but in the medium sizes the exhaust-valve stems generally work in water-cooled guides.

Few of the fuel oils in ordinary use are of such character that perfect combustion can always be attained under all conditions of load; it is therefore desirable that at least the exhaust valve, which is the most prone to fouling, may be dismantled with the greatest promptness and ease and with the least disturbance of adjustments.

More attention has been paid to the correct construction of the fuel atomizers than to any other single consideration, and with good reason. The experiments carried out on the first Diesel engine proved that the atomizing of the fuel and its introduction into the cylinder could best be performed by means of compressed air. The elements employed consist of an atomizer, an injection valve and a nozzle. These may be arranged to form the "closed" nozzle as originally developed by Diesel or the "open" nozzle as developed by Lietzenmayer. In the former the injection valve is placed between the air and fuel passages entering the atomizer and the nozzle inlet to the working cylinder, the valve thus separating both air and fuel from the cylinder until the time of injection. In the latter the injection valve is placed between the air passage entering the atomizer and the fuel inlet to the atomizer.

The "closed" nozzle construction shown in Fig. 7 is that which has attained the most widespread use. The atomizing elements consist of a series of finely perforated plates, the holes in the alternate plates being offset with reference to each other; followed by a slotted cone leading to the injection valve and to the nozzle plate at the entrance to the cylinder. Prior to the moment of injection the requisite quantity of fuel oil for the next working stroke is delivered above the perforated plates and distributes itself over these, the interior of the atomizer being in open communication with the compressed air. When the valve is lifted from its seat the air flows in a rapid stream over the plates and through the perforations, carrying the fuel with it and tearing it into fine globules. In its passage through the nozzle plate, after leaving the slotted cone, the mixture of air and oil atoms is deflected into a flat "umbrella," spreading over the surface of the piston.

The atomizer illustrated in Fig. 8 is also of the "closed" type and is the invention of Mr. Hesselmann, of the Swedish Diesel Engine Co. The typical feature of this is the siphonic arrangement of the passage in which the fuel lies between the time of its entrance into the atomizer and the opening of the injection valve, the atomizing taking place at the inner upper edge of the siphon passage.

The use of the "open" nozzle, as illustrated in Fig. 9, is confined, in America, to horizontal engines. In this the fuel is delivered by the fuel pump into a passage which, through a nozzle, is at all times in open communication with the cylinder. The compressed injection air is closed off from this passage by the injection valve. When this valve is lifted from its seat the stream of air scours over the surface of the accumulated fuel and atomizes it with an action similar to that of a file upon a metal surface. The final atomizing and spreading is performed in the passage through the nozzle.

As the load decreases and the quantity of fuel injected into the cylinder for each working stroke diminishes, the too rapid injection of the fuel

results in a more or less explosive ignition. To avoid this it becomes necessary to reduce the pressure of the injection air; that is, to slow down the injection velocity by reducing the pressure difference between the injection air and the air compressed in the working cylinder. In the cases of the "closed" nozzle this is accomplished automatically, to a limited degree; there being less fuel in the atomizer to retard the flow of air, more air is injected, and if the free-air capacity of the compressor remains constant, the injection pressure falls to correspond to this greater delivery volume. But this self-adjustment is insufficient to accomplish the purpose and has the additional disadvantage that, in spite of the more explosive ignition, the combustion is imperfect, because of the cooling effect of the excess air following the fuel into the cylinder, and it is necessary either to blow off a portion of the compressed air or to regulate the volume handled by the compressor. The latter is the general practice and is effected by throttling the suction of the compressor. This is usually done by hand, but some engines are provided with mechanical means operated by the governor.

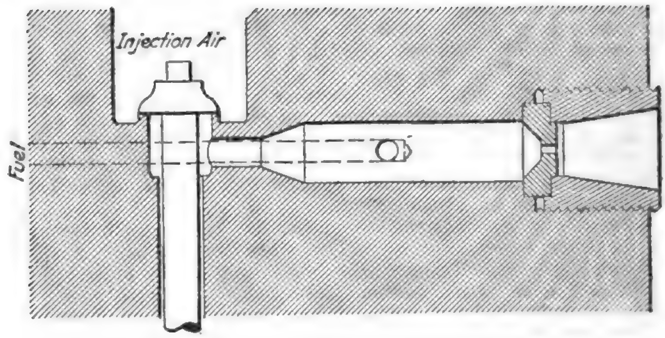


FIG. 9.—OPEN TYPE OF FUEL NOZZLE.

All engines of the Diesel type are started by compressed air. To avoid excessive use of air and to simplify the mechanism, it is usual to provide only one or two of the cylinders of a multicylinder engine with starting valves and to relieve the compression in the other cylinders until the engine has attained sufficient speed to effect this compression without risk of stopping. In small and medium-sized engines the relief of compression is generally accomplished by holding the exhaust valve partially open, which was formerly somewhat crudely done by slipping a yoke over the exhaust-valve lever; but now it is usual to mount the lever hubs on eccentric sleeves operated by hand levers, the arrangement being such that the rotation of the eccentric sleeve simultaneously places the fuel valve out of action and holds the exhaust valve open, and vice versa. This form of compression relief compels a separate handling of each cylinder and does not relieve the cylinders that are used for starting; it therefore becomes less convenient, less prompt and less effective as the size of the engine increases. For the larger sizes a system has been adopted in which auxiliary cams are used to operate the exhaust valves of all cylinders, opening these at the commencement of the compression stroke and closing them upon its completion; the fuel valves being meanwhile out of action until the engine has been accelerated sufficiently, when, by the operation of a single wheel or lever, the relief is cut off all the cylinders and the fuel is then admitted to them one at a time.

The construction of the pistons and their accessories has been reduced

to an almost uniform basis. In the larger sizes some builders provide the pistons with separate heads secured to trunk guides. Plain snap rings of cast iron have become universal, these being provided with ordinary lap joints. The piston rings are usually not doweled, to maintain fixed relative positions of the joints; except in the case of two-stroke cycle engines, in which the rings over-travel ports in the cylinder wall and it is necessary to prevent the joints from traveling across the openings. When the pistons attain diameters in which the area exposed to the heat becomes so great that radiation cannot be relied upon to prevent excessive temperatures, it is usual to positively cool the heads. This is particularly the case in two-stroke cycle engines, in which the pistons are exposed to combustion temperatures twice as often as in four-stroke cycle machines. As a cooling medium, either water, a mixture of water and air, or oil, is used.

REGULATING AMOUNT OF FUEL.

Of importance equal to that of the fuel-atomizing elements is the regulation of the amount of fuel delivered to the cylinder, to suit the existing load. The quantity, even at full load, is so minute that an almost infinitesimal variation will materially affect the speed of the engine. For example, the quantity of fuel oil delivered to a 100-H.P. cylinder, per working stroke, is only 0.0046 liter (0.28 cubic inch) at full load, and 0.0026 liter (0.16 cubic inch) at half load; or a difference of 0.002 liter (0.12 cubic inch) between full and half load. This fuel oil is delivered to the atomizers by means of a fuel pump having one plunger to serve each working cylinder; or one plunger to serve several cylinders, with a proportioning distributor to equalize the delivery to each to suit the average load conditions. Some of the methods used to regulate the fuel supply under the control of the governor are: Varying the stroke of the pump plunger; maintaining a constant stroke of the pump plunger, but bypassing back to the suction during the delivery stroke a variable quantity of the fuel; or maintaining a constant stroke of the pump plunger, with a constant pump-barrel volume during the suction stroke, and varying the pump-barrel volume during the delivery stroke.

The pressure of the air used for atomizing and injecting the fuel varies with the type of the engine and the proportionate loading. It ranges from 35 or 40 atmospheres at no load to 65 or 70 atmospheres at full load, a slight variation being desirable to suit the character of the fuel oil. This air is generally compressed by a three-stage air compressor integral with, or directly coupled to, the engine. To avoid lubrication difficulties and the danger of explosion of lubricating-oil gases, as well as to reduce the dimensions and power consumption of the compressors, they require thorough water cooling and the provision of ample inter- and after-coolers. Single-acting compressors with the trunk pistons arranged in echelon have become general, as also have automatic valves of very limited lift. As previously described, the capacity and delivery pressure of these compressors is adjusted by throttling the low-pressure suction.

LUBRICATION OF DIESEL ENGINES.

The main bearings of horizontal Diesel engines are generally constructed similarly to those of horizontal gas engines, with at least partial adjustability; whereas adjustable bearings have practically been abandoned in the modern makes of vertical engines, in favor of rigid bearings of such ample surface and provided with such efficient lubrication that their wear is negligible.

For cylinder lubrication it is desirable to use an uncompounded neutral mineral oil with a flash-point not lower than 180 degrees C. (350 degrees F.), a viscosity as high as the lubricating pump will handle reliably and

the characteristic of retaining a high degree of viscosity when hot. The function of this oil is not only to lubricate, but also to serve as a packing medium by forming a seal between the piston rings, against the leakage of air or gases. The compressor pistons are generally lubricated by a pump or by high-pressure sight-feed lubricators and use an oil similar to that for the working cylinders, except that a slightly lower viscosity and higher flash point are preferable.

In vertical engines the prevailing method of lubricating the bearings and pins is the forced-feed system, although some builders still adhere to ring oiling for the main bearings and centrifugal oiling for the pins.

To those accustomed to steam engines or, still more so, to steam turbines, the consumption of lubricating oil by a Diesel engine usually appears high, but this is the only operating expense on account of which the machine can possibly be accused of extravagance. This consumption averages 0.3 liter (0.07 gallon) per hour per 100 H.P. engine rating. Of this, about 20 to 25 per cent. is used in the cylinders.

The exhaust gases leave the cylinders at 670 to 700 degrees F. at full load, but the temperature rapidly falls with decreasing loads. To render the operation of the engine safe and comfortable, it is usual to water jacket such portions of the exhaust headers and pipes as the operator might come in contact with. Through these jackets is circulated a portion of the water that has already done duty in other parts of the engine where the rise in the temperature of the water must be maintained at a lower point. The exhaust gases from a properly designed, adjusted and operated Diesel engine are invisible and remarkably free from combustible constituents.

The majority of crude and fuel oils contain appreciable quantities of sulphur. This, if not over 1½ per cent., will not seriously affect any engine parts other than the exhaust passages and, to a less degree, the exhaust valves; but even in small amounts in combination with the hot vapors of the water of combustion, it will attack the exhaust pipes near the engine. For this reason, these pipes are made of cast iron and the use of steel for this purpose is avoided.

The noise of the exhaust is muffled by the same means as are in common use with gas engines; but concrete exhaust-muffling chambers have been found undesirable, on account of the disintegrating effect of the hot oil vapors, which may temporarily pass from the cylinders by reason of some maladjustment of the valve gear.

OILS SUITABLE FOR DIESEL ENGINES.

Fuel oils for use in Diesel engines should contain the lowest proportions of the following impurities, compatible with the prices demanded.

Water: More than ⅛ per cent. of water should be considered excessive, and if fuel containing more than this has been accepted, the water should be settled out by heating with a steam coil.

Sulphur: If in excess of 1½ per cent., the combination of the sulphurous fumes with the vapors of the water of combustion will corrode and pit the exhaust valves and seats and rapidly eat the exhaust piping.

Ash: A comparatively minute percentage of entirely non-combustible matter in the fuel causes, on the oily cylinder walls between these and the piston, an accumulation that will result in excessive wear.

Asphaltum: This much-abused term is susceptible to so many and various interpretations that it has no definite significance. Nor has the method for its determination been standardized. The various chemical and mechanical (penetration) determinations have little or no bearing upon the real issue under consideration, namely, the complete combustibility in a Diesel cylinder under the conditions existing in it and within the available time. A comparison of results obtained in actual use, with oils

containing a substance other than ash, which will not volatilize under certain definite conditions, appears to be the best guide as to the proportion of this substance that will render necessary an excessively frequent cleaning of the cylinder and its adjuncts. Several years of careful observations and records have induced the oldest Diesel-engine builders in this country to adopt, for such comparisons, the percentage of residue remaining after the sample of oil has been gradually brought to a temperature of 300 degrees C. and then subjected to this temperature for 120 hours in a closed furnace in which combustion does not take place. Under this treatment the sample is reduced to practically constant weight. It has been determined that, so long as this residue is less than 10 per cent. of the original weight of the sample, unreasonably frequent cleaning

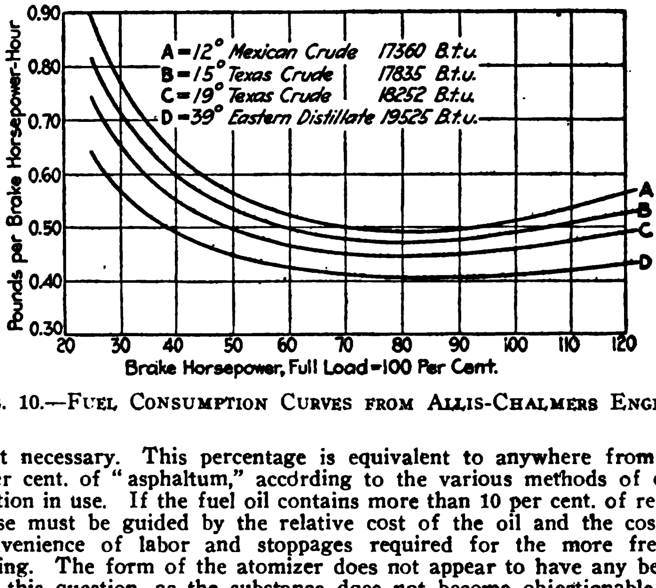


FIG. 10.—FUEL CONSUMPTION CURVES FROM ALLIS-CHALMERS ENGINE.

is not necessary. This percentage is equivalent to anywhere from 7 to 30 per cent. of "asphaltum," according to the various methods of determination in use. If the fuel oil contains more than 10 per cent. of residue, its use must be guided by the relative cost of the oil and the cost and inconvenience of labor and stoppages required for the more frequent cleaning. The form of the atomizer does not appear to have any bearing upon this question, as the substance does not become objectionable until after the fuel has entered the cylinder and its more volatile constituents have become gasified although it may render necessary warming of the fuel.

Fig. 10 illustrates the consumption of the fuel oils of widely varying gravity, in the same engine.—"Power."

TWO- VS. FOUR-STROKE CYCLE MARINE DIESEL ENGINE.

Some of the relative advantages of two- and four-stroke cycle Diesel engines for marine work and their present limitations in size and speed are contained in a paper by G. C. Davison, of the New London Ship and Engine Co., read at the recent International Engineering Congress in San Francisco. The following is in substance the author's remarks on this subject:

It is apparent that there exists today a certain difference of opinion as regards the merits of the two- and the four-stroke cycle Diesel engines.

While there are more firms building two-stroke cycle engines, at the same time those building four-stroke cycle engines seem to have met with more success. The two-stroke cycle engine superficially has the advantage of developing double the power for a given size of cylinder. It gives one impulse each revolution, while a four-stroke cycle engine requires two revolutions for one impulse. It may seem that the former should therefore furnish twice the power for an engine of given size. This, however, is far from realization. In order to operate on the two-stroke cycle principle a scavenging air system has to be provided. This requires considerable space and weight and a large number of additional working parts. The net result is that after adding the scavenging system the two-stroke cycle engine occupies nearly the same volumetric space as the four-stroke cycle engine and is only slightly lighter.

Another advantage of the two-stroke cycle engine is that with a given number of cylinders a more even turning movement is obtained. consequently a lighter flywheel can be used; in fact, with six or eight cylinders in a high-speed engine of this type, the flywheel can almost be dispensed with. Also the reversing of a two-stroke cycle engine is much simpler than that of a four-stroke cycle. The same set of cams can be used for both the ahead and the astern motion. Moreover, inertia forces in two-stroke cycle engines are masked by the pressure in the cylinders, so that stresses on the main-bearing caps and connecting-rod bolts are very small as compared with those in a four-stroke cycle engine.

The two-stroke cycle engine has no mechanically operated exhaust valves, as the exhaust occurs through ports in the cylinder wall, which are uncovered at the end of the stroke by the piston. In the four-stroke cycle type, the mechanically operated exhaust valve is exposed to hot gases and is the source of more or less trouble.

The four-stroke cycle engine has certain advantages of its own. It is simpler and each cylinder comes nearer to becoming a complete unit than in the case of the two-stroke cycle. As regards the number of parts, it has not only a less number of different parts, but also a less number of total parts, making it mechanically simpler. Also it is far superior in the heat conditions, as it handles only half the amount of heat developed in a two-stroke cycle engine of the same dimensions. Consequently, for cylinders below a certain size it does not require artificial cooling of the piston, as is necessary in the two-stroke cycle type. In the larger sizes piston cooling is necessary in both types, but the four-stroke cycle has the advantage of working with a much lower mean temperature, consequently there is less likelihood of cracks due to heat conditions. Moreover, it is more economical than the two-stroke cycle.

LIMITATIONS IN ENGINE SIZES.

There are certain limitations to the size of Diesel engines which may be advantageously applied to marine propulsion. The smallest than can well be used is in the neighborhood of 100 H.P. As compared with gasoline engines the first cost of a Diesel of small power is such that below this limit the saving in cost of fuel is sufficient only to offset the interest and depreciation. As the size increases, the economy in fuel becomes of greater importance. The upper limit of size is at present fixed by the designs and materials now in use and is approximately 400 H.P. per cylinder in single-acting units. That is, this represents about the upper limit of engines that have been built and are in successful operation. Experiments have been made, however, on double-acting two-stroke cycle engines developing 2,000 H.P. per cylinder, which represents about the theoretical limit unless some radical departure is made.

Marine Diesel engines may be roughly divided into two general classes, according to weight per horsepower. The light-weight class is repre-

sented by the engines used in submarines, which run at piston speeds from 1,000 to 1,200 feet per minute. A 1,000 H.P. 6-cylinder engine will in some cases run as high as 450 r.p.m. and will weigh approximately 40 pounds per H.P.

A heavy-duty slow-speed engine developing 1,000 H.P. at 100 r.p.m. will weigh as much as 300 pounds per H.P. Intermediate types are also built; for instance, the 1,000-H.P. Nuremberg engine of the U. S. S. *Fulton*, running at 260 r.p.m., weighs 100 pounds per H.P.

About 300 Diesel engines were built for submarines previous to the present war; since the war began a large number have been built or are building, the sizes ranging from 300 to 1,000 H.P. Destroyers require such enormous power (from 10,000 to 15,000 H.P.) that it is not practical, at least at present, to install Diesel engines to give this output. Certain foreign countries have, however, installed both steam turbines and oil engines in destroyers. For ordinary cruising the Diesel engine is used, thereby effecting great economy in fuel, and for high speed, the turbine.

The United States Navy is now building the fuel ship *Maumee*, to be equipped with two Nuremberg two-stroke cycle Diesel engines, developing about 2,500 H.P. each. These will be the largest slow-speed heavy-duty engines thus far attempted. For commercial work marine engines from 50 to 2,400 H.P. are in regular operation.

A further development, yet in its infancy, is the double-acting two-stroke cycle Diesel. Prior to the European war the Nuremberg Co. had been working for over four years on a 6-cylinder double-acting two-stroke cycle engine of 12,000 H.P. This was reported to have been completed last summer.

ECONOMIC OUTLOOK FOR MARINE DIESELS.

There has always been a difference between the European and the American point of view, due to conditions. Up to within the past year capital was comparatively plentiful in Europe and fuel scarce. Consequently the European ship-owner considered ultimate saving and was willing to pay a greater first cost for his propelling plant if the operating economy would show an ultimate gain. In the United States the shipping business has never been given much encouragement and those who have gone into the business have had to consider first costs very seriously. Furthermore, both coal and oil are comparatively cheap in this country. Finally, a further drawback to American development has been the lack of trained operators. In the course of time the basic advantages will be realized in this country and the necessary trained operators will be developed. Under present circumstances it is impossible to install a Diesel-engine plant at the same cost as a steam plant. There is a possibility that there may eventually be developed a Diesel engine which will cost so little more than a steam plant that the difference will not be worth serious consideration. When that time comes we shall probably see a boom in the marine Diesel-engine business.

Probably the worst enemies of the marine Diesel engines during the past few years have been its over-enthusiastic advocates, many of whom have made promises that they could not fulfill. Others have built and installed engines that were nothing but experiments. New firms are continually entering the field, little realizing that the design and construction of these engines are highly developed specialties. The first ones produced in this way are generally failures and unfortunately the good and the bad suffer as a result.—"Power."

HIGH-SPEED, SUPERHEATED STEAM TURBINE BLADING.

An interesting contribution to the subject of so-called non-ferrous metals and alloys used for turbine blading was given in a paper presented to the Institute of Metals on the 17th September by Mr. W. B. Parker, F. I. C., in which an exposition of the present position and of the trend of future advancement was summarized. The author also demonstrated the importance of the subject, both industrially and nationally, and advocated the scientific control of manufacture and original research.

Mr. Parker suggested that particulars of turbine designs should be interpreted into metallurgical equivalents, such as working conditions and physical and chemical properties, and showed how these are best defined in material specifications. Terms commonly used in purchasing specifications covering the physical properties were reviewed and criticised, and practical suggestions were given respecting definitions for these terms, with suitable methods for determining the properties they cover. In reviewing the types of non-ferrous materials already tried, the author stated that pure brasses of the α, $\alpha + \beta$ and β types, as well as numerous so-called manganese brasses, manganese bronzes, and other complex copper-zinc alloys, have proved unfit for modern conditions of high-speed turbines driven with superheated steam.

Copper, nickel, cobalt and the alloys of copper-nickel, copper-manganese, copper-aluminum, copper-aluminum-manganese, copper-aluminum-nickel and phosphor-bronze were critically considered by Mr. Parker, and the results of original tests made by him were given. He pointed out that, although non-ferrous materials have the advantage of non-rusting properties, at present alloy steels were utilized for the highly-heated and stressed-rotating blades because of the necessity of having a material which possesses *naturally* a good "proportional limit" which is stable, *i. e.*, not seriously lowered by prolonged exposure to highly superheated steam. None of the non-ferrous materials at present in use fulfil the latter condition, and the aim of the non-ferrous manufacturers must be, therefore, to produce a material in a physically stable condition, that is, in the naturally soft or annealed condition. This material should possess a "proportional limit" of over 16 tons per square inch, a tensile strength exceeding the proportional limit by not less than 100 per cent., and an elongation not less than 10 per cent. The proportional limit should remain constant within 10 per cent. of its "cold" value over the range of temperature from 212 degrees to 844 degrees F.

In conclusion, the author gave further particulars of a tentative specification, and schemes for researches for materials to comply with the same."—"The Shipbuilder."

HARDENING AND TEMPERING HIGH-SPEED TOOL STEEL.

THE EFFECT OF CHROMIUM AND TUNGSTEN UPON THE HARDENING AND TEMPERING OF HIGH-SPEED TOOL STEEL.*

BY PROFESSOR C. A. EDWARDS, D.Sc., AND H. KIKKAWA, MANCHESTER UNIVERSITY.

The only systematic British investigation which has been published regarding the tempering of high-speed cutting steels is that by Professor Carpenter (1, see references). The experiments which he describes were made with two series of steels—namely, chromium-molybdenum and chromium-tungsten. The latter series, consisting of five speci-

*Paper read before the Iron and Steel Institute, September 23, 1915.

mens, are all that need be considered in connection with the present work. Their compositions are given in Table I.

TABLE I.

No.	Carbon.	Silicon.	Chromium.	Tungsten.
	per cent.	per cent.	per cent.	per cent.
12	0.98	0.24	3.1	7.96
13	0.77	0.29	3.70	10.83
14	0.85	0.15	3.0	12.5
15	0.63	0.13	2.2	12.8
16	0.55	0.15	2.5	13.5

In determining the temperatures at which tempering took place, the method adopted by Professor Carpenter consisted in heating the hardened specimens for one hour at the desired temperature and then allowing them to cool in the furnace. After this treatment the sections were polished and examined under the microscope, and the tempering judged by noting the changes of structure from the polygonal austenitic crystals through the various stages towards troostite, etc., but no hardness determinations were made. Briefly, the conclusions he arrived at were as follows: "No tempering occurs below 550 degrees C.; all undergo an appreciable tempering at 550 degrees C. No. 12 is fully tempered at 600 degrees C. No. 15 at 650 degrees C., and Nos. 13, 14 and 16 at 700 degrees C." Experiments were also made to ascertain the effect of the initial hardening temperature upon the tempering of these steels, from which it was concluded that "if the alloy No. 14 be quenched below 1,100 degrees C., tempering is already well advanced at 550 degrees, and is complete at 600 degrees C.; if it be quenched at 1,150 degrees or above, tempering is less advanced at 500 degrees C., and is not complete till 650 degrees C.; but the results at 1,100 degrees, 1,150 degrees and 1,200 degress are all about the same."

Valuable as these results undoubtedly were, it soon became evident that they were inconclusive, for in discussing them along with the cooling and heating curve data which were also obtained by Professor Carpenter (2), Mr. Taylor (3) concluded that they did not provide a substantial indication of the relative cutting properties of high-speed steels. His reason for this view was that he had found, as a result of many carefully conducted experiments, that the best cutting properties were obtained after subjecting tools to a secondary heat treatment at a temperature above that for which the heating curves and microstructures seemed to show that complete tempering would occur. In consequence of this Mr. Taylor said: "These facts indicate clearly that the two methods as yet devised by scientists for determining the most important quality of the new high-speed steels are ineffective," and he was of the opinion that the only satisfactory method of deciding upon the relative merits as regards cutting was to make elaborate standard cutting-speed tests with carefully standardized tools upon standardized forgings. But he says: "This test requires so much expensive apparatus, consumes so much time, and is so slow, that a simpler index or guide which will indicate correctly the quality of high-speed tools is much needed. Moreover, we firmly believe that in time some simpler index to the property of 'red-hardness' in tools will be found."

The second heat treatment which Mr. Taylor found so beneficial consists in heating the tools, which have already been hardened, to a temperature somewhere between 370 degrees and 671 degrees C. The maxi-

mum permissible temperature for this secondary heating was not very rigidly fixed, but the following statement seems to indicate that it is somewhat lower than 671 degrees C. The second or low heat treatment consists of heating the tools—

"(*a*) To a temperature below 671 degrees, preferably to 621 degrees C., for a period of about 5 minutes; and

"(*b*) Cooling to the temperature of the air, either rapidly or slowly."

It was also pointed out that heating a little above this higher limit, even for a very short time, almost ruins the cutting properties of a tool.

From a consideration of the above facts, it is quite evident that further research upon the effect of chromium and tungsten on the tempering of modern cutting tools should be of considerable interest, and the results obtained of some practical value. Further, it would be useful to know the cause of the improved cutting properties which are brought about by the second heat treatment.

In a paper dealing with "The Function of Chromium and Tungsten in High-Speed Steels," one of the present authors (4) made a few preliminary experiments in this direction, and, judging from the evidence then obtained, it was thought that the failure of high-speed cutting tools was not due to the loss of hardness, but to the formation of a brittle constituent, which destroyed their cutting edge. It is, however, necessary to say that this conclusion is incorrect, for during the present investigation it has been found that the brittle constituent does not form on tempering steels which have been properly hardened. It can only be detected in material which has been overheated during the high heat treatment. The analysis of the steels which have been used are shown in Table II. It will be seen that these may be divided into two series: (*a*) Those with an approximately constant percentage of carbon and chromium but with varying tungsten, and (*b*) those with the same carbon and tungsten content and varying chromium. These two series are all that could be desired for determining the influence of chromium and tungsten upon the tempering properties of the steels.

TABLE II.

No.	Carbon.	Chromium.	Tungsten.	Silicon.
	per cent.	per cent.	per cent.	per cent.
1	0.63	6.15	...	0.07
2	0.65	6.13	3.08	0.05
3	0.67	6.04	5.92	0.08
4	0.65	6.19	9.12	0.04
5	0.67	6.18	12.50	0.06
11	0.63	...	19.28	0.07
12	0.63	1.12	19.40	0.06
13	0.68	3.01	19.37	0.04
14	0.64	4.91	19.33	0.09
10	0.67	5.99	18.86	0.10
9	0.64	6.24	17.69	0.07

In all cases the specimens which were used were 1 inch square and ¼ inch thick. They were hardened by heating in a crucible contained in an injector furnace to about 1,350 degrees C. for 10 minutes, followed by quenching in an air blast. In this way they were cooled to below a red heat in about 1¾ minutes. The hardened specimens were then carefully ground on a whetstone to obviate any tempering or change of structure, and finally polished for microscopic examination. If this examination re-

vealed a structure which indicated that the specimen had begun to melt, another sample was taken. In the hardened pieces the so-called austenitic or polygonal crystal structure was aimed at, but it was not possible to get this in steels with less than 3 per cent. of chromium or 9 per cent. of tungsten. The former contained numerous specks of undissolved carbide or tungstide, and the latter were more or less martensitic; in fact, No. 1, with no tungsten, was entirely martensitic.

All tempering experiments up to 800 degrees C. were made in an electric resistance furnace, the temperatures being carefully registered by means of a thermo-couple which was placed in contact with the specimens and connected with a portable pyrometer. For heating to temperatures above 800 degrees C. a gas-fired muffle furnace was used. In most cases the tempering consisted in heating for one hour after reaching the desired temperature and then allowing the pieces to cool on an asbestos pad in the ordinary atmosphere. After each treatment hardness determinations and microscopic examinations were made, and then the same specimens were heated at the next higher temperature. Preliminary experiments were first made with steel No. 9, because it contains approximately the same carbon, chromium and tungsten, as recommended by Mr. Taylor. The hardness data are given in Table III, and plotted in Fig. 1, page 1026. In this instance the hardness of the hardened material was 627, which fell to 596 after heating for two hours at 494 degrees C.; heating for 24 hours at the same temperature had practically no further effect. By heating to 542 degrees a substantial increase in the hardness is brought about, and a still further increase occurs at 589 degrees; in fact, after heating at the latter temperature the material is much harder than it was

TABLE III.—PRELIMINARY EXPERIMENTS. STEEL NO. 9. HARDENED IN AIR BLAST.

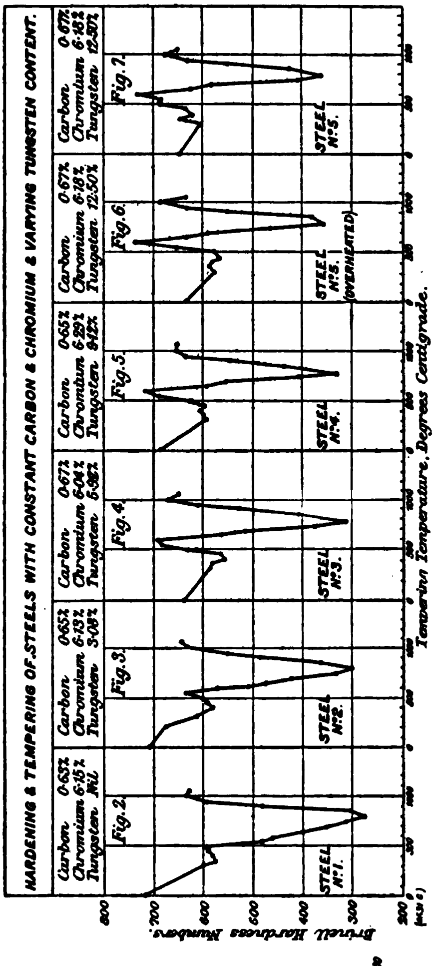
Treatment.	Indentation. Diameter. mm.	Brinell hardness No.
As hardened	2.45	627
Tempered at 494 deg. C. for 2 hours	2.51	596
494 " 24 "	2.50	600
592 " 2 "	2.41	648
589 " 2 "	2.32	700
638 " 2 "	2.36	676
657 " 2 "	2.45	627
686 " 2 "	2.56	573
711 " 2 "	2.83	467
735 " 2 "	2.92	439
784 " 2 "	3.16	373
925 " 2 "	2.48	611
986 " 2 "	2.44	632
Kept at 986 deg. C. for 2 hours, and then cooled in furnace	3.42	317

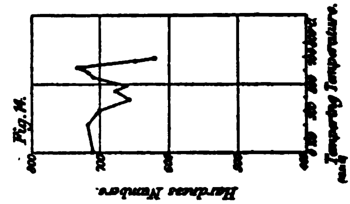
in the initially hardened state. From 589 degrees up to 784 degrees C. the steel becomes progressively softer, the Brinell figures for the two extreme temperatures being 700 and 373 respectively. When heated to higher temperatures and cooled in the same manner, the hardness is again greatly raised, and reaches 632 for 986 degrees C. Similar tests were made with this steel after heating at the various temperatures for one hour, and as the same results were obtained, this period was, with one exception, adopted for all subsequent experiments.

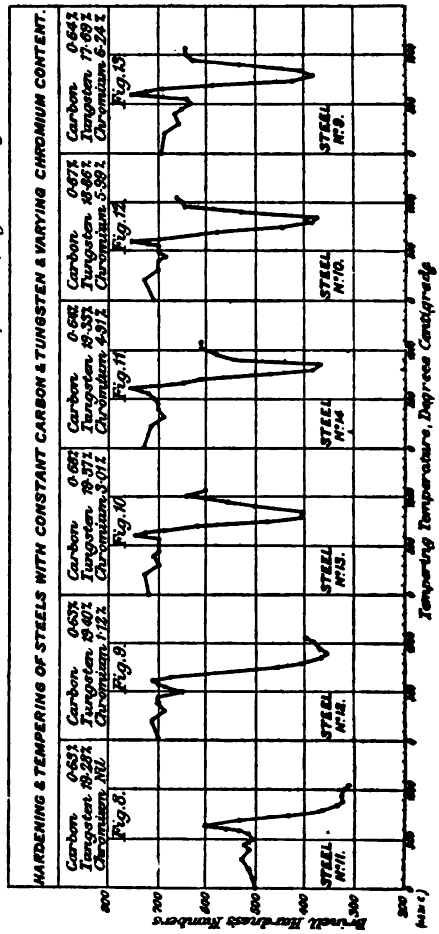
The results for steels with a constant chromium content are given*

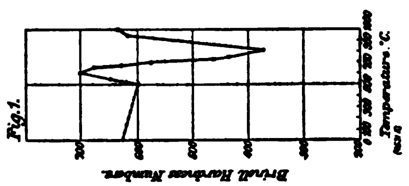
*[The demands upon our space will not permit us to reproduce Tables IV and V, but the results embodied in these tables are clearly shown graphically in Figs. 2 to 18.—ED. E.]

in Table IV, and graphically illustrated in Figs. 2, 3, 4, 5, 6 and 7; and those for steels with a constant percentage of tungsten are shown in Table V, and plotted in Figs. 8 to 13. Broadly speaking, it will be seen that the effect of tempering is very similar in all cases for the series containing the same percentage of chromium. The differences, which must be attributed to the influence of tungsten, and which are certainly









important, are clearly variations in the degree of certain features that are characteristic of all the steels belonging to this group. In all cases the first effect of tempering is to bring about a marked softening; but after heating above certain temperatures which lie between 397 degrees and 589 degrees C., they again become harder. On examining the diagrams in Figs. 2 to 7, and allowing for some slight irregularities, it will be observed that the temperature at which this secondary hardening begins is progressively raised with increasing percentages of tungsten. This rise is almost exactly 50 degrees C. for each increment of 3 per cent. of tungsten up to 12.5 per cent., and then a further rise of 50 degrees C. for the next 6 per cent. of tungsten. Perhaps the most important point to note in this connection is the total increase of the secondary hardening. With the pure chromium steel this is comparatively small; it is only from 580 to 596 in Brinell figures. It is more pronounced (580 to 636) in the presence of 3 per cent. of tungsten, and still more so with 5.92 per cent. of tungsten. In fact, the net effect in the latter case is an increase of 130 in the Brinell figure, which makes the steel much harder after being heated to 589 degrees C. than it was in the initially hardened condition. Similar results were obtained with steels containing higher percentages of tungsten, and it is significant to note that the degree of secondary hardening becomes higher as the amount of tungsten contained in the steel is raised. As regards the temperature at which the maximum secondary hardness is obtained, which also corresponds to the temperature above which real softening or annealing begins, it will be noticed that for the steel with no tungsten this is 494 degrees; for the one with 3 per cent. of tungsten, 542 degrees C.; and 589 degrees for 5.92 per cent. of tungsten; but higher percentages do not appear to have any further influence in this respect. By heating above these critical temperatures softening occurs in all cases, and the completely annealed state is reached at 784 degrees C. After heating from 784 degrees to 1,000 degrees the steels again become harder, and this is an indication of what is generally regarded as the property of self-hardening, which has usually been attributed to the action of tungsten; but the data contained in Fig. 2 very clearly show that a pure chromium steel also possesses this property to a very remarkable extent.

Turning now to the results obtained from the series with a constant percentage of tungsten, which are illustrated in Figs. 8 to 13. The main features for those steels containing more than 3 per cent. of chromium are very much the same as those just described. Marked differences are, however, evident for the pure tungsten steel and the one containing 1.2 per cent. of chromium. In the former case the hardness after the high heat treatment and air-quenching is only 500, which is much lower than for the pure chromium steel. Tempering up to 494 degrees C. has very little influence, but from 494 degrees to 638 degrees C. the Brinell figure is raised to 610. With 1.2 per cent. of chromium the hardness falls only very little after heating up to 638 degrees C. It is a most curious fact that in both these cases the temperature at which real softening begins is 50 degrees higher than in any of the other steels, but it is necessary to remember that in the one without chromium the hardness never reaches such a high value as with those containing chromium. From 638 degrees to 884 degrees C. there is a rapid decrease in the hardness; but whereas in all other cases heating at from 884 degrees to 1,048 degrees C. is followed by a marked degree of self-hardening, in the pure tungsten steel this property is entirely absent under these conditions of cooling, and it is not very pronounced in steel No. 12.

In view of the fact that Mr. Taylor recommends a five minutes' heating for the secondary treatment, it was thought advisable to determine the effect of that period as compared with the results obtained after an hour's heating. This was done in the case of steel No. 9, and the results are

shown in Table VI and Fig. 14, from which it will be seen that the only difference is in the temperature at which the maximum degree of secondary hardening is obtained. After heating for five minutes, this temperature is 25 degrees C. higher than on tempering for one hour.

TABLE VI.—TEMPERING FOR FIVE MINUTES.

Tempering temperature. Deg. C.	Brinell hardness No.
As hardened.	708
207	719
302	700
397	659
445	676
494	665
542	708
589	719
614	732
638	708
662	648
686	627

As a result of the foregoing experiments it is possible to offer a more rational explanation of the function of chromium and tungsten in modern high-speed tool steels. Contrary to the opinion which has been held by some workers, it is now unquestionable that the element chromium imparts to steel the property of self-hardening by cooling in air from high temperatures. Indeed, a steel containing 6 per cent. of chromium, along with 0.63 per cent. of carbon, possesses this property in a much greater degree than one containing 18 per cent. of tungsten; it is not only much harder after the usual air-quenching from about 1,300 degrees C., but its hardness is progressively raised, even with comparatively slow rates of cooling, as the initial temperature is increased from 800 degrees to 1,000 degrees C. On the other hand, no hardening occurs when the tungsten steel is cooled under exactly the same conditions from those temperatures.

Briefly, then, the action of chromium in these steels is as follows:

(*a*) In conjunction with carbon, it is the cause of the great hardness of high-speed steels; and

(*b*) It produces a marked lowering in the temperature at which hardening can be effected.

As regards tungsten, its action in the absence of chromium is to raise the temperature at which tempering or annealing begins, and in the presence of chromium it increases the intensity of the secondary hardness which is brought about by the low heat treatment, and raises the tempering temperature.

There now appears to be very little difficulty in giving a trustworthy explanation of Messrs. Taylor and White's important discovery of the fact that a secondary low heat treatment improves the cutting efficiency of modern high-speed steel. When this treatment is conducted at the temperature they advise—namely, 620 degrees C.—it gives the maximum degree of hardness. Although in the authors' experiments the hardness was determined at the ordinary temperature, after the heating, there seems to be no doubt that the figures obtained in this way are a measure of what Mr. Taylor describes as "red-hardness." For the determinations which were made after heating for periods of five minutes, the maximum hardness obtained corresponds to 614 degrees C., which is practically the same temperature as Mr. Taylor suggests. These facts lead the authors to believe that valuable information as regards the relative merits of high-speed cutting-steel can be obtained from careful tempering experiments when made in conjunction with hardness determinations. Fur-

ther, when more data are accumulated this simple method may be found sufficient to replace almost completely the very elaborate and costly standard cutting tests which are now necessary. In any case it should be extremely useful for ascertaining the best temperature for the second or low heat treatment.

Effect of Initial Hardening Temperature.—Mr. Taylor has made it abundantly clear that the hardening temperature is a factor which has a most decisive influence upon the cutting properties of high-speed tools. He found that the higher the temperature, short of actual melting, the greater the cutting efficiency. This improvement is certainly not due to an increase in the hardness of the steel, because whenever differences are observed in this connection it is invariably found that a greater hardness is produced by air-quenching from the lower temperatures down to about 1,000 degrees C. Cooling curves which have been taken from temperatures between 1,000 degrees and 1,320 degrees C., as shown by Professor Carpenter (1) and one of the present authors (4), indicate that the initial temperature has a pronounced effect upon the character of the carbide change-point, but it would really be going too far to say they are sufficient to show that better cutting powers should be obtained by treating the tools from the higher temperatures. Hence with a view of getting more direct and decisive evidence respecting this aspect of the question, tempering experiments were made with steels which had been hardened by air-quenching from about 1,050 degrees C. Several steels were used for this purpose, but as the results were practically the same in each case, it is only necessary to refer to a typical example. The data given in Table

TABLE VII.

Tempering temperature. deg. C.	Indentation. Diameter. mm.	Brinell hardness No.
As hardened	2.25	744
302	2.35	683
397	2.38	665
445	2.39	659
494	2.39	659
542	2.37	670
589	2.51	596
614	2.64	535
638	2.73	503
686	3.01	415
735	3.22	359
784	3.29	343

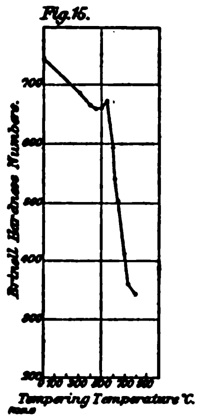
VII and plotted in Fig. 15 relate to steel No. 9. It will be noted that the hardness was in the first place 744, which is considerably greater than for the same steel when quenched from just below its melting-point, 1,350 degrees C. After tempering for an hour at 300 degrees C., the hardness fell to 683, and at 445 degrees C. to 659; it then remained the same for 494 degrees C., and increased slightly at 589 degrees C., but rapidly decreased at all higher temperatures. When compared with the tempering curve (Fig. 13) it will be readily seen that these results are in entire agreement with Mr. Taylor's discovery. They show that, although the steel is harder when quenched from 1,054 degrees C. than from 1,350 degrees C., it loses its hardness much more readily, and only gives a faint indication of the secondary hardening. This difference is very marked, for with the lower hardening temperature annealing begins at 542 degrees C., whereas in the other case the maximum hardness is obtained at 589 degrees C., and at those temperatures the respective Brinell figures are 670 and 756.

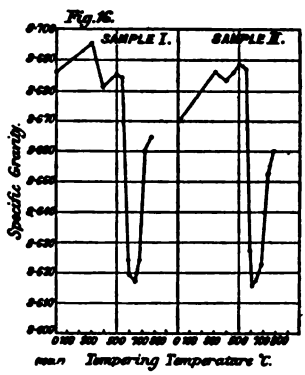
It is now possible to give a clear idea of why the tempering properties of these steels are so largely dependent upon the hardening temperature, but for this purpose it is necessary to remember the following facts:

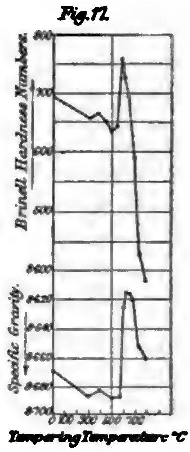
1. The whole of the carbon and chromium of a steel containing carbon 0.63 and chromium 6.15 per cent. is in complete solution at all temperatures above the Ac critical points; and

1*a*. These elements can be retained in solution by air-quenching from any temperature above the carbide change, and when this is done the steel is as hard as water-quenched carbon steel.

1*b*. A hardened chromium steel of the kind under discussion only develops a comparatively slight degree of secondary hardening (see Fig. 2, page 1026), and the Brinell hardness drops from 720 to below 600 after the steel has been tempered at all temperatures above 300 degrees C.





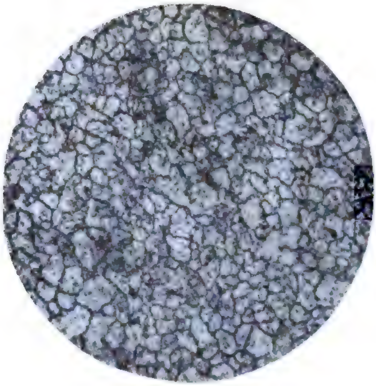


2. With a steel containing about 19 per cent. of tungsten and 0.63 per cent. of carbon the latter element goes into solution immediately the Ac points have been passed, but a very large proportion of the tungsten remains undissolved even when the temperature is raised to the melting point. This is shown in Photomicrograph No. I. It is not known in what form this undissolved tungsten exists, but it is most probably present as the compound Fe_2W, which has been isolated by Professors Arnold and Read (5).

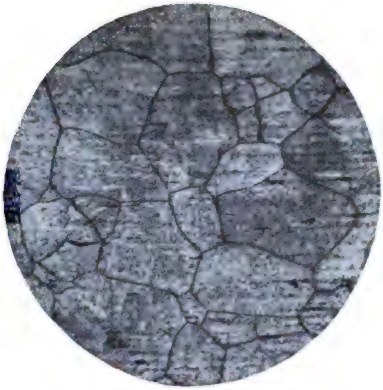
2*a*. Very rapid air-quenching from temperatures up to 1,050 degrees C. has practically no hardening effect upon the above tungsten steel, and even when quenched from about 1,350 degrees C. it is much softer than a similar specimen of chromium steel which had been slowly cooled in air. (See Table VIII.)

2*b*. But when tungsten steel is tempered it undergoes a very pronounced secondary hardening, and, in fact, after being heated to 638 degrees C., it is harder than the chromium steel which has been tempered at any temperature above 300 degrees C.

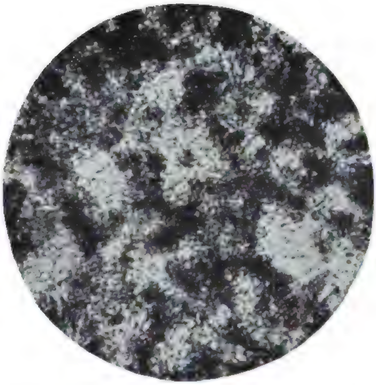
3. The presence of chromium in high-speed steel increases the solubility of the tungsten to such an extent that with 6 per cent. of the former element, 19 per cent. of the latter can be held in solution at about 1,350 degrees C.



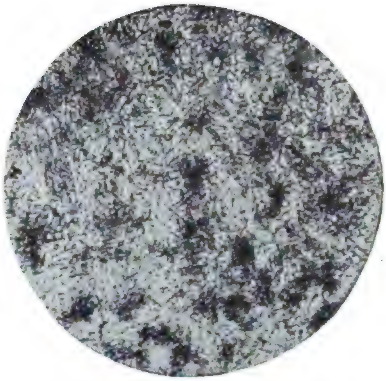
No. I.—Chromium, 1.12 per cent.; Tungsten, 19.14 per cent. Air-quenched from 1350 deg. Cent. Magnified 150 diameters and slightly reduced.



No. II.—Steel No. 9. As hardened. Magnified 150 diameters and slightly reduced.



No. III.—Steel No. 9. Tempered at 494 deg. Cent. Magnified 150 diameters and slightly reduced.



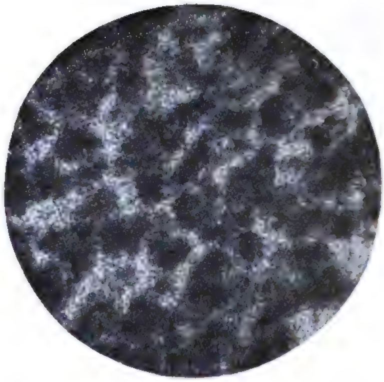
No. IV.—Steel No. 9. Tempered at 589 deg. Cent. Magnified 150 diameters and slightly reduced.

Nos. I to IV.—Tempering of High-Speed Steels.

No. V.—Steel No. 9. Tempered at 638 deg. Cent. Magnified 150 diameters and slightly reduced.

No. VI.—Steel No. 9. Tempered at 784 deg. Cent. Magnified 150 diameters and slightly reduced.

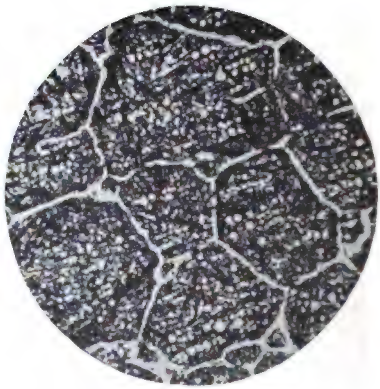
No. VII.—Steel No. 9. Tempered at 884 deg. Cent. Magnified 150 diameters and slightly reduced.



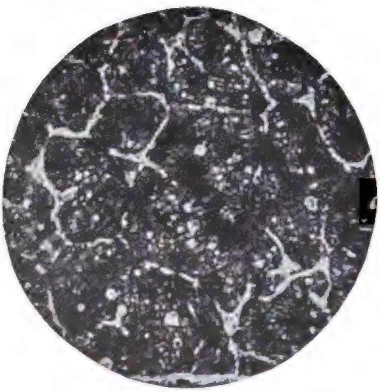
No. VIII.—Steel No. 9. Tempered at 986 deg. Cent. Magnified 150 diameters and slightly reduced.

Nos. V to VIII.—Tempering of High-Speed Steels.

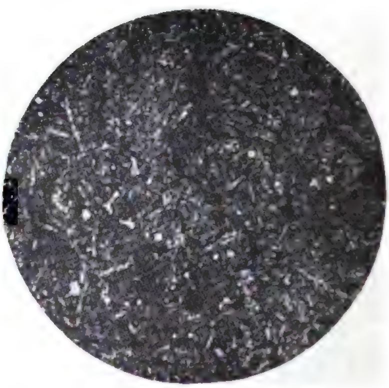
No. IX.—Overheated during hardening. Magnified 150 diameters and slightly reduced.



No. X.—Tempered at 302 deg. Cent. Magnified 150 diameters and slightly reduced.



No. XI.—Tempered at 445 deg. Cent. Magnified 150 diameters and slightly reduced.



No. XII.—Tempered at 784 deg. Cent. Magnified 150 diameters and slightly reduced.

Nos. IX to XII.—Over-Heated and Tempered High-Speed Steel.

TABLE VIII.—HARDNESS OF STEELS NOS. 11 AND 9 AFTER COOLING IN AIR FROM DIFFERENT TEMPERATURES.

Size of specimens, 1 inch by 1 inch by ⅞ inch.

...	Temperature.			
.	764 deg. C.	834 deg. C.	914 deg. C.	986 deg. C.
	Brinell No.	Brinell No.	Brinell No.	Brinell No.
No. 11	268	282	286	313
No. 9	377	464	622	652

...	Temperature. 1058 deg. C.	Air-quenched from 1048 deg. C.
	Brinell No.	Brinell No.
No. 11	313	354
No. 9	655	676

Bearing the above facts in mind, the authors believe that the influence of initial temperature may be correctly summarized as follows:

1. When the hardening temperature is low the tempering properties approximate to those of a pure chromium steel, softening takes place at lower temperatures, and little or no secondary hardening occurs. Under these conditions the full influence of the tungsten is not obtained, because a large proportion remains inert in consequence of it being out of solution.

2. The maximum resistance to tempering, and the greatest degree of secondary hardening, can only be obtained by getting the tungsten in complete solution, and for modern high-speed steels this can only be effected by heating the steel to about 1,350 degrees C.

Microstructures.—Since the microstructures of these steels are in many ways so remarkably similar, and as their chief features have been previously described (2 and 4), it is only necessary to refer to the structural changes that occur after they have been tempered at the various temperatures. For this purpose all that is needed is to describe the structures of one steel, which may be taken as typical of the series. No. II (facing 1030) represents the structure of steel No. 9 in the hardened state, and shows the well-known polygonal austenitic crystals. The structure of the same material after being tempered at 494 degrees and 589 degrees C. is quite martensitic (Nos. III and IV); this corresponds to the appearance of the secondary hardening. After being heated to 638 degrees C. (No. V), specimens are more readily etched, and show signs of the decomposition of the martensitic structure, but the martensitic markings are still evident, and are not completely removed until a temperature of between 700 degrees to 750 degrees C. is reached. After being heated to 784 degrees C. the material becomes fully annealed, but the structure No. VI shows irregular-shaped patches of carbide. When heated to 884 degrees C. and cooled, hardening again sets in, but the structure No. VII is so remarkably granular and fine as to make it impossible accurately to describe it. There can, however, be very little doubt that it is partly martensitic, because at

the next higher temperature it becomes almost purely martensitic. (See No. VIII.)

Steels which have been overheated or partly melted show unmistakable signs of this treatment. The structure of a specimen of this kind is shown in No. IX, where the broad crystal boundaries and globular patches in the interior of the crystals represent portions of the steel which were liquid at the temperature from which they were air-quenched. From Nos. X, XI and XII, it will be seen that the white background passes through the same structural changes on being tempered as a properly hardened specimen, but the partly melted portions are very persistent, and not entirely removed after heating for an hour at 784 degrees C. Steels which have been overheated are invariably brittle, and are liable to crack under the Brinell test, in which case the fracture follows the crystal boundaries.

Hardness and Specific Gravity.—When metals are hardened, either by mechanical work or by heat treatment, changes of volume usually occur, and considerable importance is now attached to these changes, because they are supposed to indicate the cause of the hardening. The present paper would therefore be incomplete without some reference to this question.

For this purpose it was thought that the best results would be obtained by first hardening the steel, then, after carefully grinding and polishing each surface of the specimen, determine its specific gravity, and repeat these determinations on the same samples after it had been tempered at the various temperatures for an hour. The object of using the same sample was to eliminate all possible errors that might otherwise be introduced by slight variations in the composition, or other defects, in different samples. Two independent series of experiments were made with steel No. 9. The results that were obtained are given in Table IX, and plotted

TABLE IX.

Tempering temperature.	Specific gravity.	
	Sample No. 1.	Sample No. 2.
Deg. C.		
0	8.686	8.670
302	8.695	8.686
397	8.681	8.683
445	......	8.685
494	8.685	8.688
542	8.684	8.687
589	8.624	8.627
614	8.619	8.616
638	8.617	8.617
686	8.624	8.622
735	8.660	8.652
784	8.664	8.660

in Fig. 16. The chief characteristics of the two curves are fundamentally the same; indeed, with the exception of the values in the hardened state and those after tempering at 300 degrees C., the results are virtually identical. In considering the details of these results, it will be seen that the effect of tempering is first to increase the density or specific gravity of the mass; at 397 degrees C. there is a slight decrease, which, in turn, is

followed by another small increase at 445 degrees, 494 degrees, and 542 degrees C.; but the most striking discontinuity occurs between 542 degrees and 614 degrees C. This change corresponds with a very remarkable decrease in the density which takes place at precisely the same temperature range as that of the secondary hardening. Above 614 degrees C. —that is, when the steel begins to be annealed—the density is again raised. When these results are compared with the tempering hardness curve for the same steel, Fig. 13, the authors venture to say that the most casual observer cannot help being impressed by the resemblance which undoubtedly exists between the graphical illustrations of the two sets of experiments. Indeed, it is almost impossible to resist the conclusion that there is a direct connection between hardness and volume. This will perhaps be more evident on examining Fig. 17, where the density and hardness data are reproduced in such a way as to give diagrams of approximately the same size. From these curves it will be seen that each increase of hardness is accompanied by an increase in volume, and vice versa. The authors do not propose at this stage to utilize these extremely interesting facts to support any particular theory of hardening steel. One of the reasons for doing this is that they are quite consistent with at least two of those theories. The essential points to remember are:

1. The greatest hardness is associated with the highest volume.

2. When metals are hardened by mechanical deformation, similar volume changes are obtained.

Conclusions.—The conclusions which have been drawn from this investigation are:

1. The first effect of tempering hardened high-speed steels is to make them softer, but when they are tempered at higher temperatures they again become harder, and after heating at or about 614 degrees C., they are much harder than in the initial air-quenched state. There can be no doubt that this secondary hardening is the cause of the improved cutting powers of a tool, which Mr. Taylor found was brought about by a secondary low heating to about 620 degrees C.

2. Chromium in conjunction with carbon is the cause of the great hardness of hardened high-speed steels, and further, it materially lowers the temperature at which these steels can be air-hardened.

3. In the absence of chromium, tungsten raises the temperature at which tempering or annealing begins, and in the presence of chromium it increases the intensity of the secondary hardening, and raises the tempering temperature.

4. Tungsten steel containing 18 per cent. of tungsten and 0.63 per cent. of carbon can be air-hardened only by rapid air-quenching from temperatures above 1,050 degrees C.

5. When high-speed steels are hardened at low temperatures, say 1,050 degrees C., the tempering properties approximate to those of a pure chromium steel, by softening at a low temperature, and developing little or no secondary hardening. This is due to the tungsten being undissolved and remaining inactive.

6. The maximum resistance to tempering and the greatest degree of secondary hardening can only be obtained by getting the tungsten into solution, and with modern high-speed steels this is not complete until a temperature of about 1,350 degrees C. is reached.

7. Specific gravity determinations seem to indicate that there is a direct connection between the hardness and volume of these steels. On tempering, an increase of hardness is accompanied by an increase in volume.

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HIGH-SPEED REDUCTION GEARS.

During the past week we have had an opportunity of seeing under test a set of reduction gears for a steam turbine, just completed for Messrs. Workman, Clark & Co., of Belfast, by the Power Plant Company of West Drayton, Middlesex. This firm may not unfairly claim to be the pioneers in this department of engineering, having cut the gears supplied to the *Vespasian*, which demonstrated the possibilities of the system. The gears for Messrs. Workman, Clark & Co. are designed to transmit 2,200 horsepower with a reduction ratio of 10 to 1. The pinion is of 7.66-inch pitch diameter, and has twenty-three double helical teeth, and runs at 3,000 revolutions per minute. The wheel is of 77.33-inch pitch diameter, with 229 teeth, the useful width of the face being 3 feet, and the width over all 4 feet 6 inches. From the foregoing, it appears that the pitch-line speed is 100 feet per second, and the pressure 336 pounds per lineal inch of effective width of wheel. The test was made running light, the pinion shaft being driven by a 30-horsepower motor. The Power Plant Company have, we should add, supplied the plant complete, inclusive of its housing, oil pump, oil filter and oil cooler. The total floor space occupied measures 9 feet 2 inches by 9 feet 10 inches, and the center line of the shafting is 2 feet 9½ inches above ground level. The gear ran under test with very little noise. The lubrication is forced throughout, the oil being supplied under pressure by a small centrifugal pump arranged at one end of the pinion shaft. All the bearings are accessible without removing the main cover. The brasses are lined with white metal. The supply of oil to the pinion enters the center of a long sleeve extending from end to end of the pinion, and cut away on one side so as to clear the teeth of the driven wheel. At each end this sleeve is bored to fit the external diameter of the pinion for a length (axially) of about 1 inch; but the intervening space is larger than the pinion diameter, and this annular space is kept filled with oil by the pump. A uniform distribution of lubricant over the whole length of the teeth is thus secured. The pinion is of nickel-steel specially heat treated. The teeth of the driven wheel are cut in forged-steel rings shrunk on to cast-iron center-pieces, machined all over and accurately balanced. The above gear, large as it is, by no means represents the full capacity of the works. A set recently sent out weighed 25 tons complete. The teeth of these gears are hobbed on special tools designed and constructed at the works. The largest of these machines is capable of cutting wheels up to 13 feet 6 inches in diameter by 80 inches width of face. It is, however, probable that the Power Plant Company may ultimately abandon the hobbing process for these large gears in favor of a method of "generating" the teeth. Small double-helical wheels are already being made with generated teeth, and

with such excellent results that a large machine operating in the same way is now under construction. The new machine is a very ingenious one and has the advantage that the dividing gear has to take very much smaller pressures than when the teeth are formed by means of a hob.—"Engineering," October 1, 1915.

THE SURFACE CONDENSER.*

BY C. F. BRAUN.

The primary functions of a surface condenser are to reduce back pressure on the exhaust side of a steam prime mover; to conserve and return to the steam generator, in the water of condensation, as many units of heat as possible, and to remove air from the feed water, thus reducing pitting of boilers. In accomplishing these results the condenser must handle four separate fluids, viz: steam, air (including other non-condensable vapors), water of condensation and cooling or circulating water. The desirable state of these fluids is not the same, the conditions to be approached being:

(*a*) *Steam* should enter the condenser, be conducted freely to all parts with the least possible resistance, reduced to the lowest practical temperature (and consequently pressure), and converted into water.

(*b*) *Air*, a non-conductor, should be rapidly cleared from the heat-transmitting surfaces, collected at suitable places, practically freed from entrained water and water vapor, and cooled to a low temperature for removal at minimum volume, with consequent least expenditure of mechanical energy.

(*c*) *Condensate* should also be rapidly cleared from the heat-transmitting surfaces, freed from air, collected at suitable points for removal, and returned to the steam generator at the maximum practical temperature.

(*d*) *Circulating water* should pass through the condenser with least friction, deposit a minimum amount of precipitated chemicals or debris, and absorb a maximum amount of heat.

The main principle of the design of the condenser is the transference of heat from the steam through the dividing surface to the cooling water, and the transfer per unit of area or of size is a measure of the efficiency of the apparatus, this being directly proportional to the temperature difference or head. For obtaining mean values of temperature differences the following formula, developed mathematically by Grashof, has been proved accurate:

$$M = \frac{D_1 - D_2}{\log_e \frac{D_1}{D_2}}, \qquad (1)$$

where

M = mean temperature difference,
D_1 = temperature difference between fluids at beginning,
= $TS_1 - TW_1$,
D_2 = temperature difference between fluids at end,
= $TS_2 - TW_2$,
TS_1 = initial temperature of steam,
TS_2 = final temperature of steam,
TW_1 = initial temperature of circulating water,
TW_2 = final temperature of circulating water.

It is commonly assumed that TS is constant throughout the condenser, by which this formula reduces to

$$M = \frac{TW_2 - TW_1}{\log_e \frac{TS - TW_1}{TS - TW_2}}, \qquad (2)$$

* Abstract of paper read before the Spring Meeting of the American Society of Mechanical Engineers, June, 1915.

but since the frictional drop through the steam space of a condenser is usually 0.5 inch or more, representing with high vacuums a temperature difference of, say, 10 degrees F., the use of formula (2) for applying to large condensers data obtained on smaller ones, or for analyzing the performance of a condenser or various sections of a condenser, will lead to serious errors. With any given set of temperature values the mean temperature difference can only be varied by arrangement of heating surfaces. These must be such as to produce counter-current flow, the circulating water entering where the steam is coolest and leaving where it is hottest.

Transfer of heat through a unit of condenser-tube area per unit of mean temperature difference was early recognized as varying greatly under different conditions, the most apparent variation being an increase with increase of water velocity. With regard to resistance the transfer of heat produced by the temperature head is opposed by a total resistance R, which for analysis divides conveniently into the resistance R_v on the vapor or steam side of the surface, the resistance R_m of the metal walls of the surface, and the resistance R_w on the cooling-water side of the surface. A simple equation expressing heat transfer in useful terms may be written as follows:

$$R = \frac{M}{H},$$

in which

H = number of heat units transferred per unit time,
M = mean temperature difference,
R = total resistance $= R_v + R_m + R_w$.

Even with high steam pressures and with superheat, the total B.t.u. to be extracted by the condenser may safely be assumed as 1,000, and it is convenient to adopt an arbitrary resistance unit such that

$$H = \frac{1{,}000 \times M}{R} \text{ or } W = \frac{M}{R}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

in which

H = B.t.u. per square foot per hour,
M = mean temperature difference in degrees F.,
R = resistance per square foot of surface,
W = pounds steam condensed per square foot per hour.

The symbol U will be used when M is unity, so that U = B.t.u. per square foot per hour per degree mean temperature difference. This resistance R may also be expressed in terms of equivalent conductivity by the equation:

$$R = \frac{1{,}000 \times L}{C \times 4{,}290},$$

in which

L = thickness of substance in inches,
C = conductivity in c.g.s. units

The results of the tests by Orrok, as shown in Fig. 1, curve A, provide the most reliable data available on heat transfer through condenser tubes; and in Fig. 2, curve A, are plotted total resistances R obtained by applying the values from Orrok's curve in Fig. 1 to equation (3) in which M is taken as unity. These resistances are plotted against reciprocal velocity instead of against velocity. A reasonable value for $R_v + R_m$ for Orrok's test is 0.4, and on this assumption curve B, Fig. 2, is plotted showing the relation R_w to the reciprocal velocity.

A curve, Fig. 1, curve B, plotted from the values on curve B, Fig. 2 represents the relation of heat transfer from the surface of a condenser tube to velocity of the water in contact with that surface. From this curve U_w varies directly with V according to the equation:

$$U_w = 245 + 141\, V.$$

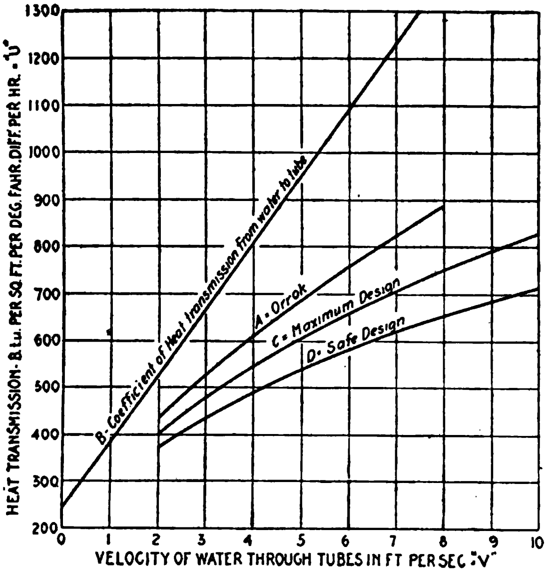


FIG. 1.—HEAT TRANSMISSION—VELOCITY CURVES.

This variation of resistance, inversely with velocity, is due to the fact that the particles of water in contact with the surface at any instant form a non-conductor which prevents the flow of heat from particles in the body of the water to the surface of the tube. The transfer of heat is really by convection, and the more rapid the removal of the heated particles, and their replacement by cooler ones, the greater the heat transfer. With the same velocity this transfer of particles is much more rapid in a small tube than a large one, where, so to speak, a cold core of water exists. The desirability of small tubes is thus indicated, and experience shows that a ¾-inch to ⅞-inch diameter should be a maximum.

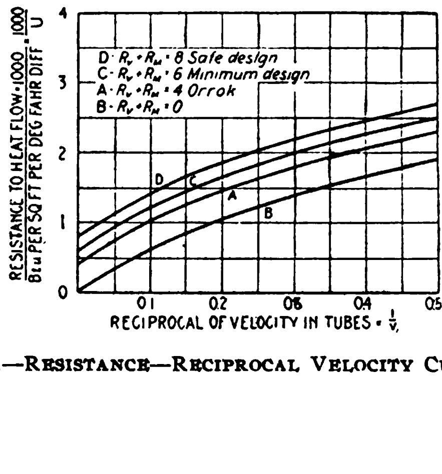


FIG. 2.—RESISTANCE—RECIPROCAL VELOCITY CURVES.

A well-designed cylindrical condenser is illustrated in Fig. 3. Most condensers consist of a cylindrical shell containing closely-spaced round tubes, the water passing through the tubes and the steam around them. Practically all condenser tubes are made of copper or a high-percentage copper alloy, and the size of the diameter of the tube is the determining factor in its thickness. The arrangement of heating surfaces for easy cleaning, and the construction of water-channel covers independent of pipe connections, are important. To prevent the formation of a coating of oil, which has an effect more serious than a coating of scale, a fairly high steam velocity must be maintained over the tubes, and corners which become stagnant must be eliminated. An exhaust opening of liberal size with a dome extending the length of the shell, as in Fig. 3, will cause the steam to be distributed to the ends of the tubes and prevent stagnant corners.

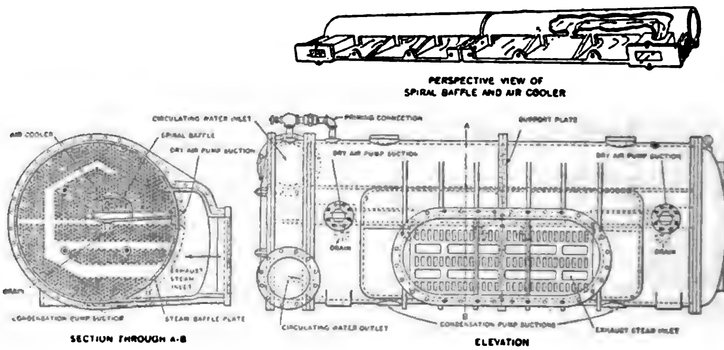


FIG. 3.—CYLINDRICAL CONDENSER OF GOOD DESIGN.

Baffle plates for directing the steam to remote parts of the condenser introduce resistance to steam flow and should be avoided, except in the case of the small plate directly in front of the exhaust inlet which protects the tubes from entrained water in the exhaust. The steam passing over the tubes condenses and diminishes in volume as it progresses, but the steam-flow velocity may be maintained by constructing a gradually reducing steamway, or by gradually reducing the pitch of the tubes, or by making lanes or passages to various parts of the steam space by omitting tubes (see Fig. 3). If a liberal pitch be employed for the tubes and ample lanes be provided, the frictional drop need not exceed 0.4 inch. It is important that a sufficient number of air-removal connections be located at points where air accumulates. Even distribution of water through all tubes is necessary, and narrow channels causing high velocities or inlets directing water on to the tubes must be avoided. Steam should be maintained at the lowest practicable pressure, and the temperature must approach closely that of the circulating-water discharge, although a difference, which should not be more than 10 degrees F., must be kept in order to produce heat flow. An air cooler is an essential part of a condenser.

Regarding general structural features, proper provision for the accommodation of expansion strains may be accomplished by expanding the tube into one head and packing at one end only. The proper supporting of tubes to prevent sagging and cracking is important, and supporting plates drilled true with the tube sheet should be located not more than 8 feet apart. Shells should be made of cast iron, and not of steel, which is corroded by the gases contained in the steam, and water channels and covers should be separate castings, so that the tube ends may be readily exposed for cleaning without

breaking the pipe connections. The connections between, and relative location of, condensers and auxiliaries are important factors in condenser efficiency, but in most cases are beyond the control of the manufacturer and consequently are neglected, thus becoming a constant source of trouble. The exhaust pipe, condensate piping, and air-pump piping should all be amply large, although the last is ordinarily much larger than necessary.

A comprehensive rating of the surface condenser must include the following:

(*a*) Quantity of steam condensed.
(*b*) Vacuum obtainable (corrected to 30-inch barometer).
(*c*) Temperature of available cooling water.
(*d*) Cooling-water exit temperature.
(*e*) Condition of air at point of removal.
(*f*) Friction head on cooling water.
(*g*) Temperature of condensate at point of removal.

The first four items express the heat-transmitting efficiency, and can, for purposes of comparison, be reduced to B.t.u. per square foot per degree difference per hour.

Since a higher vacuum at the air pump than at the exhaust inlet is of no value, the mean temperature difference used for comparing results on condensers should be computed on the assumption that the steam temperature throughout the condenser is that due to the vacuum at the exhaust inlet, and equation (2), the values of which can be determined from Fig. 4, should be used.

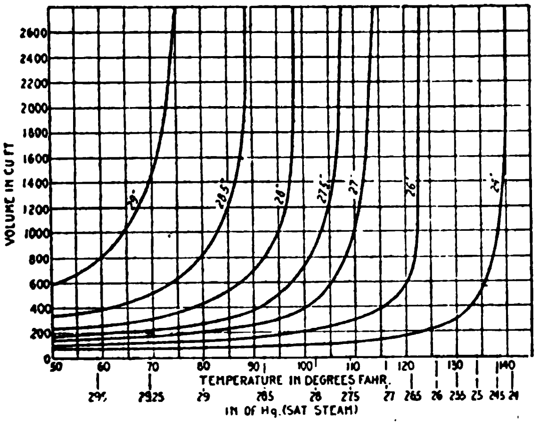


FIG. 4.—CURVES SHOWING THE VOLUME PER POUND OF AIR MIXED WITH SATURATED WATER VAPOR, FOR VARIOUS TEMPERATURES AND PRESSURES OF THE MIXTURE.

A diagram for the rapid determination of mean temperature difference values is given in Fig. 5. Probably the most reliable formula for determining these is that of Grashof, which, when the temperature of one of the mediums is constant, reduces to

$$D = \frac{T_2 - T_1}{\log_e \dfrac{T_s - T_1}{T_s - T_2}},$$

where D equals the mean temperature difference, T_1 the lowest temperature of the fluid, T_2 the highest temperature of the fluid, and T_s the temperature of

the gas. The result will not be changed if any constant be deducted from all the T's in the equation. By making this constant T the equation is simplified to one of three variables, which are plotted on the diagram, Fig. 5. To increase the accuracy or range of the diagram two scales are given: for

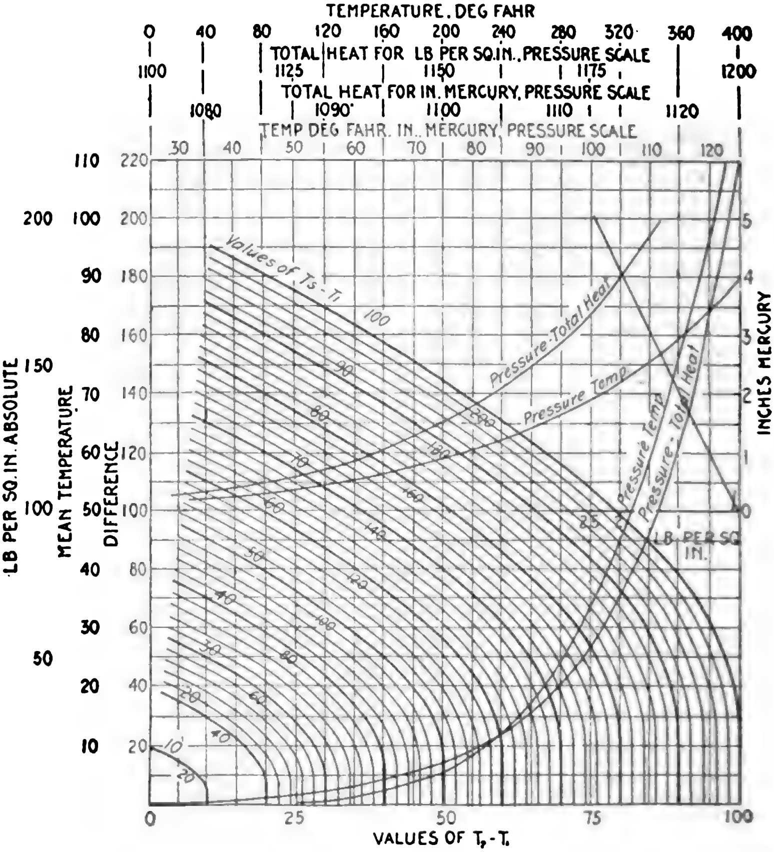


FIG. 5.—CURVES FOR SOLUTION OF EQUATION [2].

temperatures between 200 degrees and 1,000 degrees use the 100 scale. Five curves showing pressure-heat temperature relations of steam are produced on the same set for convenient reference.

The complete equation for the condenser is

$$S = \frac{W \times Q}{M \times U} \text{ or } U = \frac{W \times Q}{M \times S},$$

in which

U = B.t.u per square foot per degree F. difference per hour,
= coefficient of heat transmission,
M = Mean temperature difference in degrees F.,
W = Pounds of steam condensed per hour,
S = Square feet of cooling surface,
Q = Total heat removed by circulating water per pound steam condensed (usually taken as 1,000 in all cases for simplicity).

TABLE I.

	A	B
Square feet surface	2,000	5,500
Pounds steam condensed per hour	18,500	57,200
Vacuum, inches	28½ (1½ abs.)	27
Temperature entering cooling water, degrees	65	60
exit cooling water, degrees	80	100
Mean temperature difference per equation (1), degs.	18.3	30.9
B.t.u. per square foot per degree difference per hour	505	337

Thus, as an example, if it be assumed that the results in Table I are obtained from condensers A and B, A will be 50 per cent. more efficient than B. Items 3 to 6 in Table I determine the mechanical energy required by air condensate and circulating pumps, and item 7 indicates the heat efficiency, being a measure of the heat returned to the system in the condensate.

—"The Shipbuilder."

SALVAGE OF THE *F-4*.

NAVAL CONSTRUCTOR J. A. FURER.

Up to the time the *F-4* was lost, the United States Navy had been entirely free from serious submarine accidents. In fact, our Navy was the only one operating any considerable number of under-water craft which had not lost one or more such vessels.

The four submarines of the "F" class had been operating in Hawaiian waters since August, 1914. On the morning of March 25th, 1915, at about 9:15, the *F-4*, preceded by the *F-1* and *F-2*, stood out of Honolulu Harbor for a submerged run. On the way out, the *F-4*, running submerged, but with periscopes showing, passed the *F-1*, standing in on the surface. The commanding officer of the *F-1*, noticed that the periscopes of the *F-4* were trained on him and waved his cap in acknowledgment. That was the last signal exchanged by the doomed vessel with the living world.

The other two submarines returned to the harbor shortly, but when the *F-4* had not returned by 10:30 A. M., the officer of the deck of the tender *Alert* became alarmed and reported the fact to the commanding officer. A fast motor boat was promptly dispatched to search the harbor fairway for any signs of the vessel. The other three submarines and all the power boats of the *Alert* presently joined in the search. In the meantime, also, men had been stationed to listen in on the submarine signal sets of the *Alert* and of the other vessels. This watch was kept for days, but not the slightest sign of life was ever received from the lost submarine.

At about 12 o'clock one of the vessels came across quantities of air bubbles and an oil slick on a range approximately between Diamond Head and Barber's Point. The chart shows 1,200 feet of water in this locality. Had this spot been the resting place of the *F-4*, there would have been no use in attempting to bring her to the surface. However, the oil seemed to come from inshore, which left some hope that the vessel might not lie beyond the depth of human assistance.

The ocean floor rises very steeply to form the Island of Oahu, as it does around all of the islands in the Pacific Ocean which are of volcanic origin. Only a few hundred yards inshore from the locality covered by oil, soundings gave a depth of about 300 feet. This was encouraging and left a chance that the vessel might not be in the very deep water indicated by the first bearings.

Two divers from the submarine flotilla made a descent in this vicinity

to a depth of 190 feet without reaching bottom and without seeing anything of the *F-4*. These men went down under helmets only—this is a matter of choice, because of the somewhat greater facility in coming to the surface in case of an accident to the air supply. This depth was later exceeded by one of the divers who descended to 215 feet under a helmet alone, but again without reaching the bottom or seeing the vessel. A descent to this depth, with air supplied by a pump, is probably a record. The return of this diver without suffering from acute caisson disease or other physical impairment is almost miraculous.

The only possible chance of saving any lives—if the men survived the accident at all—was to drag the craft into shallower water. This was known to be feasible only in case the living and machinery spaces of the submarine had not been filled with water. With the vessel completely waterlogged, her weight would have been more than 260 tons. This was no doubt actually the condition of the submarine within a few hours after the accident. There was, of course, not the slightest possibility of dragging any such weight in shore and up hill. To have made no attempt at dragging would, however, have meant giving up all hope of saving the lives of the crew at once, because no lifting gear could be improvised and made fast at a depth of 300 feet, within the time available for rescuing the men.

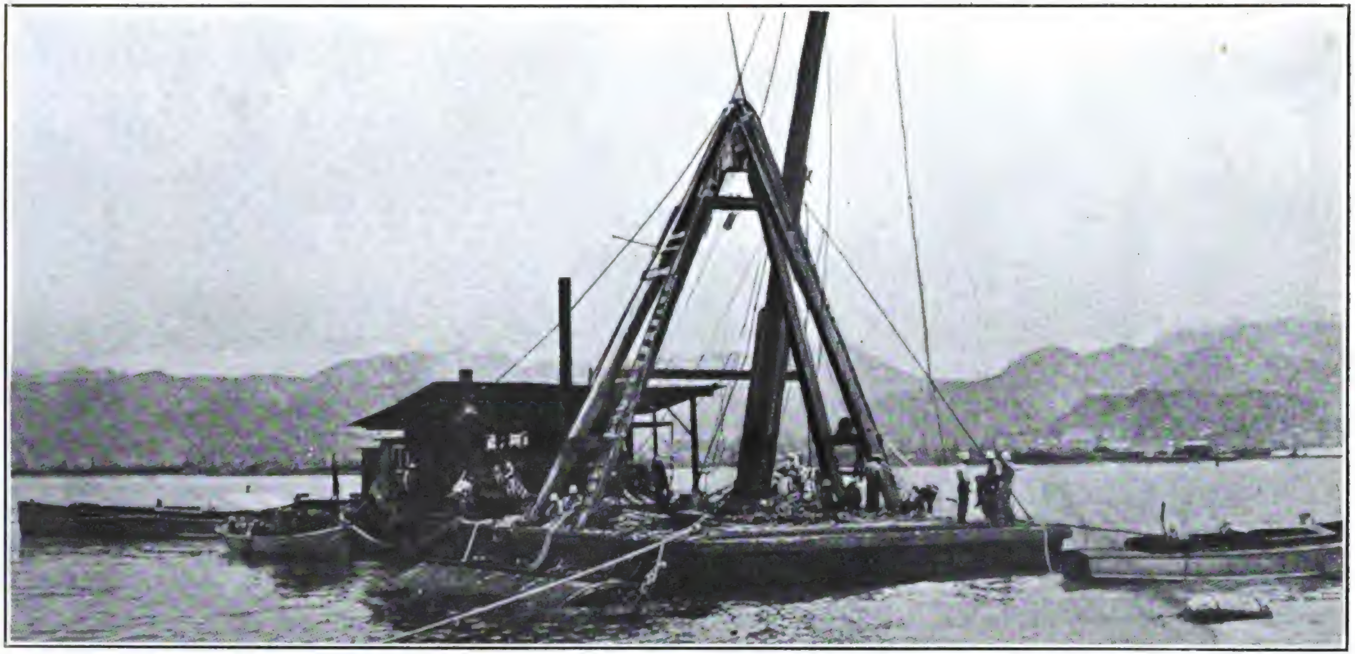
Tugs were brought to the scene of the accident to drag the bottom. For this purpose, a sweep about 2,000 feet long was made of chain and wire cables. Two tugs towed at each end, parallel to each other, and about 400 feet apart. On account of the great depth of the water, a very long sweep had to be used to insure its dragging the bottom. It was difficult to determine, even approximately, the location of the submarine, because the air bubbles showed only intermittently and the oil slick soon spread over a very large area.

During the night a microphone was constructed on the *Alert* for sounding the bottom and detecting the vessel in case of contact. This apparatus consisted of two sounding leads strapped together. Into the bottom of each lead a nail was driven. Electric conductors were connected to the nails and fitted to a telephone receiver. A number of dry cells were wired in to complete the outfit. On bridging the two nails with a metallic object, the circuit through the telephone receiver was completed—recording a click.

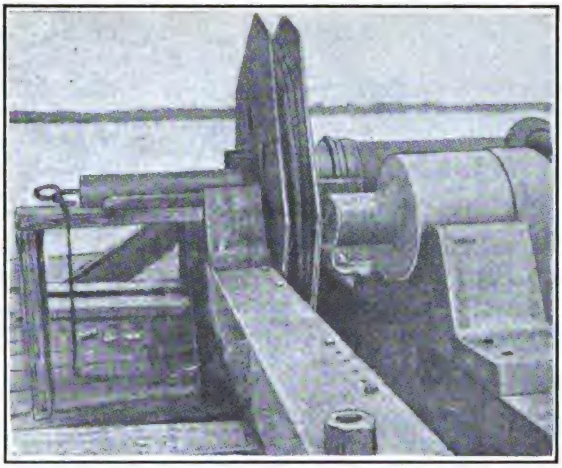
After dragging the sweep back and forth over the bottom all night, it caught firmly under the vessel about noon the next day. A good many strikes had been made previously, some of which may have been around the top of the submarine, but slipped off. The only evidence that the vessel had been caught was that air bubbles again appeared in the locality where they had been previously noted. Also, by sounding with the microphone over this area, a large metallic object was located which was correctly assumed to be the submarine. The depth of water around this spot varied from about 295 feet to 315 feet. The position was plotted and the vessel found to be lying about a mile and a half from the harbor.

All efforts to drag the vessel in shore were, however, fruitless. Even after applying an upward pull of about 100 tons, by means of a dredge, and towing in shore with all the power available, the submarine could not be moved. This was the final confirmation of the belief that the vessel must be completely filled with water. While these attempts at rescuing the crew were being made, steps were being taken to salvage the vessel by lifting. No ready-at-hand salvage gear of any description was to be had in Honolulu. This was not, however, a particular handicap, because none of its usual salvage methods could have been applied in any case, on account of the great depth of water in which the vessel lay.

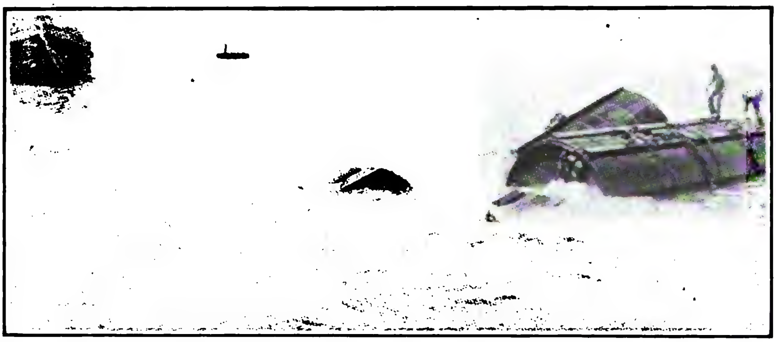
For the main units of the salvage plant, two substantially-built mud scows, 104 feet long by 36 feet beam, were selected. The use of the



Pontoon Alongside the Wrecking Scow Ready to be Sunk.

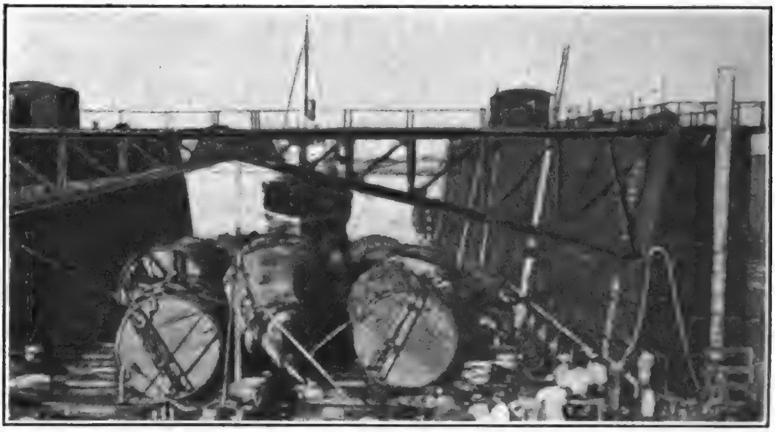


Winding Spools of Windlass—Locking Pin at Left.

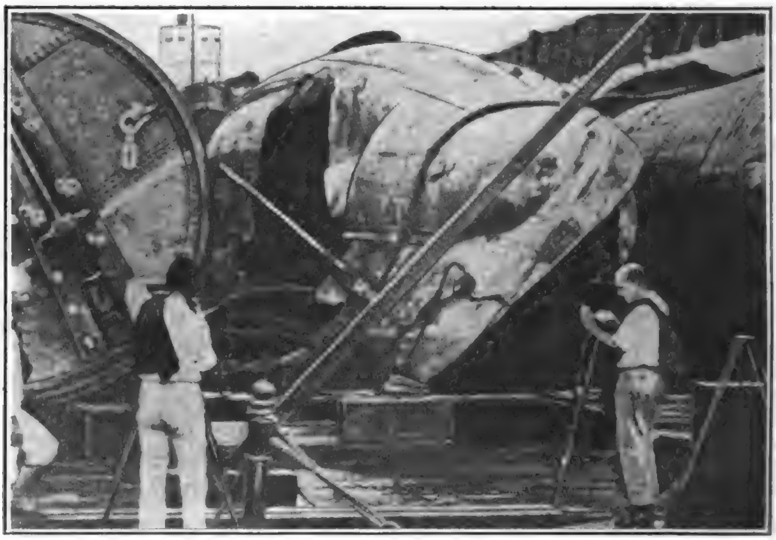


F–4 Coming Up, Bow Pontoons on Surface, Middle Pontoons Just Emerging.

Towing the F–4 Suspended From the Pontoons.



THE F-4 AND PONTOONS IN DRY DOCK.



THE LARGE HOLE TORN BY THE SLINGS THROUGH THE SHELL PLATING.

scows strictly as pontoons—that is, the employment of their buoyancy alone for lifting—was ruled out, because there is practically no tide off Honolulu.

Windlasses were improvised instead, for lifting. Five 30-inch I-beams were first of all laid across the mud pockets so as to transfer the large concentrated loads to the fore-and-aft framing of the scows. Fortunately, heavy shafts could be procured in Honolulu, as they are used in the sugar mills, and are consequently carried in stock by the machine shops. Shafts 13 feet long by 14⅝ inches in diameter were selected for the windlasses—the lifting slings being wound up directly on the shafts. These shafts were supported by three cast-iron pillow blocks secured to the I-beams. Two windlasses were fitted side by side on each scow so that two slings might be used under each end of the vessel. It would have been quite impossible to obtain a single wire hawser of sufficient strength to carry one end of the submarine under the live-load conditions to which the slings were subjected.

A hoisting engine was installed on each scow for winding up the slings. These engines were, naturally, not powerful enough to do the work without a large purchase. In the case of one scow, the power had to be multiplied 18 to 1, and, in the case of the other—which had a smaller engine—36 to 1. This was accomplished for each windlass by reeving up a triple-sheave tackle with ¾-inch wire rope leading to one of the engine drums. To the traveling block of the tackle the winding spool rope (as shown in one of the photographs) was shackled. This rope was wound up by hand on a spool which was driven on to the square end of the windless shaft. By unwinding this rope by power, the requisite turning effort was applied to the windlass shaft. As soon as the two blocks of the triple purchase came together, the spool rope was unshackled from the traveling block—rewound on the spool by hand, the triple purchase overhauled, and again shackled to the rope, ready for another lift. During this fleeting process the windlass shaft had to be kept from turning backward. This was accomplished by slipping a nickel-steel locking pin through a hole in the spool—as shown in one of the photographs—and backing down until the ends of the pin rested on pillow blocks provided for the purpose. Four locking-pin holes were bored in each spool.

The construction of the winding spools was one of the most bothersome features of the design, because of lack of manufacturing facilities. A steel-casting plant was not available. Cast iron could not be trusted, as the stresses to which the spools were subjected in service were very great. They were therefore built up solid of steel plates. The core was made of ten ½-inch plates in the absence of any heavier available material, the side plates being of ¾-inch material. All were riveted together and the core lined with Douglas fir. As the diameter of the spools was 50 inches and the diameter of the windlass barrel 15 inches, a multiplication of power of better than 3 to 1 was obtained.

The ends of the hoisting cables were secured by means of hook bolts passing through holes in the windlass shafts. These bolts were 2½ inches in diameter. By setting up on the nut at one end of the bolt, the hook was drawn down over the cable so hard that friction prevented the end from slipping through. The end was also backed up by cable clamps and three dead turns of the lifting hawser allowed before taking up the load. The length of the scows limited the travel of the running block of the triple purchase to about fifty feet. This consequently limited to three turns the number of revolutions of the windlass shafts obtainable without rewinding the spool rope. The windlasses then had to be locked, the blocks overhauled and the rope rewound. This was the condition on the scow having the more powerful engine. On the other scow, on account of the introduction of a luff to increase the purchase, only one-half turn of the shaft could be obtained between fleets. While this method of

lifting promised to be very slow when the design was evolved, it developed to be an insignificant item in the total time consumed in the salvage operation. On one occasion the submarine was lifted 58 feet in less than two hours after all the preliminary work of preparing to lift had been completed. This preliminary work took 21 days.

In designing the gear, shafting and supports, the live-load conditions due to the almost continual swell had to be reckoned with. If the assumption could have been made than 260 tons of dead load would be distributed equally among four lifting slings, the design could have been much simplified. Instead of that, the assumption had to be made that half the weight of the vessel would come as a live load on a single sling at times. That this assumption was not extravagant was fully demonstrated during the salvage work.

Two steam-driven pumps were installed on each scow, so that in case it should become necessary to use more power for starting the submarine off the bottom than could be delivered by the hoisting engines the buoyancy of the scows could be called on. The buoyancy of the scows had to be called on only once, however.

A dredge, from which the boom and bucket gear had been removed, was used as the central unit of the salvage plant. This dredge was selected because it was equipped with a 10 x 14-inch engine geared to six drums, any of which could be operated independently of the others. This dredge was anchored near the submarine. After sweeping the lifting hawsers under the submarine, the ends were taken to the dredge and hove in by means of the drum wires—thus mooring the dredge over the vessel. The gearing between the engine and the drums was such that a very considerable pull could be applied to the hawsers, as was frequently necessary in order to make the lines render under the hull of the submarine to the desired position.

Search was made, while the scows were being rigged, for hoisting cables of requisite strength to carry the loads. Cables of these dimensions are not carried in stock by manufacturers, and, as was to be expected, had to be picked up here and there. Only one cable could be located which was entirely satisfactory. This was a plough-steel wire 2⅝ inches in diameter. For the other three slings galvanized steel cables 2½ inches in diameter, and plough-steel 2 inches in diameter had to be used. One of the chief difficulties in handling hawsers of such size is their weight and stiffness. The 2⅝-inch plough-steel cable weighed over seven tons and was not much more flexible than a bar of iron.

Within a few days after the accident the Navy Department started a party of divers from New York for Honolulu to assist in the salvage operations. The Bureau of Construction and Repair had been conducting experiments for some time on deep-sea diving. Even though the divers could do very little work at a depth of 300 feet, their services were needed to report on the position of the lifting hawsers so that they could be shifted intelligently to the desired locations.

Contrary to the general belief, a special suit is not worn for deep-sea work. The diver is subjected to the full pressure corresponding to the depth in which he is working, as is the case in ordinary shallow-water diving. The difference between the deep-sea procedure and the shallow-water procedure lies merely in the method of supplying air, and in the scientific restoration of the diver's body to a normal state of equilibrium while returning to the surface after having been subjected to abnormal pressures. Air for deep-sea work is supplied from storage tanks—the pressure being regulated from above, according to the depth at which the diver is working. In this instance a number of torpedo flasks were charged to a pressure of 2,250 pounds. The air is supplied to the diver's helmet through the usual hose line after being passed through a reducing valve. The diver also has control valves so that he can regulate the air

supply. By means of the helmet valves he can make himself lighter or heavier at will. When the submarine was in 300 feet of water the pressure to which the diver was subjected when on the bottom was about 135 pounds to the square inch.

The rate at which the diver is brought to the surface is fixed by the length of time he was exposed to any given pressure. For example, the first diver to go down on this job made the descent in five minutes and was on the bottom at a depth of 306 feet for twelve minutes. He was then brought back to the surface in one hour and forty-five minutes to insure gradual decompression.

The diver is allowed to come up fairly rapidly until a depth of 100 feet is reached. A Jacob's ladder of this length, with rungs spaced 10 feet apart, is sent down to him so that when he reaches this depth in his ascent he can rest on the lowermost rung. The diver is required to stop on each rung for the length of time called for by certain tables which have been worked out from experimental data. Decompression is in this manner accomplished slowly enough so that he will arrive at atmospheric pressure restored to normal condition.

Although the diver does not become conscious of fatigue while working under great pressure much sooner than if he were working near the surface, the exertion is, nevertheless, much more exhausting. This was demonstrated in the case of a diver who remained on the bottom 30 minutes, trying to pass a small reeving line—the only actual manual work at this depth which was attempted. According to the diver's statement, he was not conscious of becoming fatigued while on the bottom, but after coming to the surface he collapsed from exhaustion and did not regain his strength for three or four days, notwithstanding the fact that the man was a perfect physical specimen.

After innumerable difficulties the four wire hawsers were placed under the submarine in the desired locations and the ends made fast to the windlass shafts. As was feared, however, the strength of the cables became impaired rapidly, owing to chafing, where they passed around the bilges of the vessel. After lifting about 25 feet and towing inshore to a new landing one of the smaller hawsers parted. While it was being replaced bad weather set in, and presently all of the other hawsers parted likewise. Just one month from the date of the accident not a line was around the vessel and the only measurable progress was a reduction in the depth of water from 300 to 275 feet.

However, the adequacy of the lifting gear had been demonstrated as well as the feasibility of sweeping lines under the vessel. It was now obvious that wire hawsers could not be counted on to resist the chafing action caused by the more or less continuous swell. Four slings were made up instead, of 90 feet of 2⅝-inch chain into the ends of which the wire hawsers were spliced. By having only chain in contact with the hull of the vessel the stranding of the wire hawsers was avoided. Chain causes more damage to the plating of a vessel when so used than wire. For this reason that type of sling was ruled out originally. Great difficulty was experienced in getting the new slings in place, as the former method of sweeping by means of two tugs was impracticable, but they were finally worked to the desired locations.

The vessel was lifted 225 feet in a few days, once the slings were in place, and towed inshore to a spot just off the channel entrance where the depth of water is only 50 feet. Preparations were being made for a final lift which would have brought the submarine high enough to permit entering the channel and landing the vessel in a floating dock in the harbor. Only a few hours were needed for this work, but in that short space of time a heavy ground swell set in and wrecked the operations. High surf waves built up in an incredibly short time which caused the scows to charge back and forth so violently that the shell plating of the

submarine gave way under the forward slings. All the lines had to be let go in order to save the vessel from becoming a complete wreck and the scows from being washed up on the reef. The ground swell was followed shortly by heavy southerly weather. When, several days later, the sea had moderated sufficiently, divers were sent down to examine the condition of the vessel. A large rent was found in the hull forward where the chains from the leading scow had ripped through the shell plating. Lifting by the windlass method was now unsafe, as it was likely that the forward end of the submarine would break off under the strain.

A new plan was then adopted consisting of the use of six specially constructed pontoon cylinders. All of the cylinders were 32 feet long—four being 11 feet in diameter and two 12 feet 6 inches in diameter. The combined lifting capacity of the six pontoons was 420 tons. A margin of 160 tons was allowed for breaking the vessel away from the bottom, should there be considerable sand suction, and for the failure to realize all of the buoyancy available which would be occasioned by some of the cylinders becoming cockbilled to such a degree as to make it impossible to blow all of the water out of the high ends.

Each cylinder was divided at midlength by a transverse, water-tight bulkhead so as to facilitate control in sinking. A 4-inch flood and discharge valve was installed in each end bulkhead close to the bottom. Air valves were fitted on top of the cylinders for blowing out the water and venting each compartment. Eight feet from the ends, 12-inch hawse pipes were fitted through the pontoons.

The plan called for lifting the vessel by means of six chains—the ends of the chains being brought up through the hawse pipes of the pontoons, three of which were to be placed on each side of the vessel. The chains were first of all worked under the submarine in predetermined positions—two forward, two amidships and two aft. The two chains of a pair were spaced 16 feet apart so as to correspond to the distance between the hawse pipes. The pontoons were planted by means of a wrecking scow which was moored accurately over the submarine. The two cylinders of a pair were towed to the scene of operations and placed one on each side of the scow, directly over a pair of chains lying on the bottom.

The ends of the chains were now fished up through the hawse pipes and the cylinders submerged by opening the flood valves. They were kept under control while sinking by means of 5-inch manila lines made fast to the ends and taken to the hoisting engines on the scow. The process therefore consisted of threading the pontoons on to the chains. After they had landed on the bottom the chains were adjusted so as to leave the necessary amount of slack to permit the pontoons to rise just clear of the vessel on becoming buoyant. A clamp was now fitted to each chain by the divers, just above the hawse pipe. The clamps consisted of two steel castings, molded to the shape of the chain links and of such length that they spanned the hawse pipes. By means of four heavy bolts the two halves of the clamp were drawn together. These bolts kept the clamps from spreading and the chain from slipping through when subjected to the lifting strain.

After all six cylinders had been landed on the bottom alongside of the vessel and the clamps had been secured by the divers, the unwatering process was started. For this purpose torpedo air flasks, charged to a pressure of 2,150 pounds, were placed on a coal barge. The flasks were piped to an expansion chamber which in turn was connected to a manifold. Twelve ¾-inch air-hose lines were led from the manifold valves to the blow-out valves on top of the pontoons. The coal barge was moored about 50 feet to one side of the location of the submarine so as to be clear when the latter rose to the surface. A pressure of 35 pounds was used at the manifold for blowing out the water while the cylinders

were resting on the bottom. The unwatering operation had to be watched very carefully so as to avoid the danger of blowing out the heads of the cylinders on emerging.

The method proved successful in every detail. The work of placing the chains under the submarine was started on August 21st. All of the chains were in position on August 25th. It then took one day to place each pair of cylinders. This could be done only during daylight, as, naturally, the divers could not work in the dark. On August 29th everything was ready for blowing the water out of the pontoons. This operation took about two hours from the time the air was turned on. One end of the submarine came up slightly ahead of the other, as was to be expected. The vessel was towed into the harbor suspended from the six pontoons, and was docked the following day.

The pontoon method which was finally adopted could not have been used originally for bringing the vessel to the surface because of the great depth of water in which the submarine lay at first. The divers had to work under water from five to nine hours a day from August 21st to August 29th while placing the chains under the vessel, securing the clamps after the pontoons had been lowered, closing flood and vent valves, and, later on, connecting the hose leads for blowing out. This work could not have been done at a depth of 300 feet, as most of it was extremely arduous and fatiguing.—"Scientific American."

GALVANIC CORROSION DAMAGES HULL OF YACHT.

A very large new sailing yacht, the *Sea Call*, built for Alexander Smith Cochrane, of Yonkers, N. Y., at a cost said to have been in the neighborhood of half a million dollars, has been so injured by corrosion, within three months after her launching, that she has been dismantled and scrapped. The case is of especial interest to engineers, as it is one of the most remarkable instances on record of serious damage resulting from the galvanic action of dissimilar metals in a structure exposed to sea water.

The yacht was intended by its owner for ocean cruising. He desired to visit out-of-the-way corners of the world where no facilities exist for dry-docking or hauling vessels out of the water to remove marine growths from the hull. It was therefore decided to build the portion of the hull under water of monel-metal plates, using steel plates, however, above the water line. Monel metal is an alloy produced and marketed by the International Nickel Co. and named in honor of Ambrose Monell, its president. It is an alloy containing about 67 per cent. nickel and 27 per cent. copper. As might be expected from these chief constituents, the metal is highly resistant to corrosion. It has a strength about equal to that of mild steel.

According to the statement of the builders, it was the original intention of the vessel's designer, William Gardner, to use monel metal for the stem of the vessel, the rudder frame and the propeller frame, as well as for the under-water plating. Some difficulty was found in producing these parts in monel metal with the necessary accuracy and freedom from warping, and it was finally determined to use steel for these pieces instead.

The framing of the hull was of steel bulb angles. Monel-metal rivets were intended to be used in all monel-metal plates. In a very few cases, however, steel rivets were used by mistake in the monel-metal underwater plating. Attention was first attracted to the serious corrosion that was going on by the failure or total disappearance of one of these steel rivets. A stream of water entering through the open rivet hole was temporarily stopped by a pine plug and later by sending down a diver

who had put in a steel bolt as a permanent repair. In a very few weeks, however, this bolt was also eaten away. About this time it was decided to haul the yacht out of the water to remove the marine growths that had accumulated on the hull, for it was found, contrary to the expectatoin of the designers, that the salt water had so little effect on the monel-metal plates that the hull rapidly accumulated enough "grass" to check the vessel's progress. The serious corrosion which had taken place on the steel parts under water was discovered when the hull was exposed to view.

The most severe corrosion took place on the thin steel rib which forms the outer frame of the rudder (the rudder itself being covered with monel-metal plates). The reason why the corrosion was so severe here was very likely because of the rapid flow of water across the rudder edge, removing the corrosion as fast as it took place and thus presenting clean metal to the action of the electric current. A similar effect, almost as severe, took place on the steel stem of the vessel. Deep pits were eaten into the steel. There was heavy corrosion also on the steel propeller-frame casting and rudder shank.

The monel metal itself, as would be expected, showed not the slightest evidence of corrosion anywhere; and had the corrosion been confined to these outer steel pieces described, they could probably have been replaced at no serious expense by similar parts of monel metal and the galvanic action thus terminated. A more serious situation, however, was presented by the interior of the vessel. It was clearly evident, from the action on these exterior steel parts, that as the interior steel framing was connected to the monel-metal plates, corrosion would be inevitable there under the action of the bilge water. The probability that this might proceed unnoticed until the vessel's structure would be seriously weakened was doubtless the consideration that led to the decision to cut up for scrap a vessel which had been launched no longer ago than last March.

The *Sea Call* was 150 feet long on the load-water line, with a length over all of 214 feet, her bow having an enormous overhang. The beam of the vessel was 33 feet 6 inches and the draught was 18 feet. The vessel had a heavy lead keel covered with monel plates, and in addition a large centerboard. It was to be equipped with a 400-H.P. internal-combustion engine as auxiliary power, but the engine was never installed.

The yacht was dismantled at the yards of her builders, George Lawley & Sons, at Neponset, Mass. On Tuesday, August 24, the hull was hauled out of the water on the marine railway at the Lawley yards, where it was inspected by one of the editors of "Engineering News."

The possibility of galvanic corrosion occurring had evidently suggested itself to the designers of the vessel, for we are informed that experiments extending over several months were made at the Lawley yards on plates of monel metal and steel electrically connected and immersed in sea water. These experiments, we are informed, showed no appreciable corrosion of the steel and no measurable current in the wire connecting the plates, and it was therefore deemed safe to proceed with the construction of the *Sea Call* on the lines described. The present condition of the vessel, however, is indisputable evidence that the conditions in the boat itself were entirely different from those which existed in making the experiments referred to. It is doubtless true that the area of monel metal exposed to the sea water being very large in comparison with the area of steel exposed, the corrosion of the steel was concentrated.

It seems indeed strange that some such action should not have been foreseen in view of the large amount of experience on record as to the effect of galvanic action between steel and some of the copper alloys when immersed in sea water. Monel metal, we are informed, is almost identical with manganese-bronze in its electrical relation to steel when connected as a galvanic couple. In ships which use manganese-bronze propellers it

has become common practice to attach plates of zinc to the hull. As the zinc is more electro positive than the steel, the corrosion due to the galvanic action with the manganese-bronze propeller is concentrated upon the zinc and the steel is protected.

Only a few miles distant from the Neponset shipyard is the Charles River dam. On the lock gates of this dam truck axles of manganese-bronze were used. In "Engineering News" of March 18, 1909, Edward C. Sherman described the extensive tests which were carried on to ascertain the amount of galvanic action that would take place on steel and manganese-bronze joined electrically and immersed in sea water. As a result of these tests the steel portions of the dam were protected from corrosion by wrapping zinc sheets around the bronze axles.—"Engineering News."

NAVAL VESSELS.

UNITED STATES NAVAL VESSELS UNDER CONSTRUCTION.

DEGREE OF COMPLETION.

No.	Vessel.	Building yard.	Engines.	No. shafts.	Speed, knots.	Percentage machinery completed 1915. Oct. 1	Percentage machinery completed 1915. Nov. 1	Percentage hull completed Nov. 1, 1915. Total.	Percentage hull completed Nov. 1, 1915. On ship.
	BATTLESHIPS:								
36	*Nevada*	Fore River S. Co.	Curtis turbine	2	20.5	96.25	96.98	97.4	97.3
37	*Oklahoma*	New York S. Co.	Reciprocating	2	20.5	96.85	96.90	98.2	98.2
38	*Pennsylvania*	Newport News Co.	Cur. trb. grd. cr.	4	21	83.39	85.03	90.4	88.1
39	*Arizona*	Navy Yard, N. Y.	Pars. trb. grd. cr.	4	21	57.70	60.49	74.6	72.5
40	*California*	Navy Yard, N. Y.	Electric	4	21	....	...	...	...
41	*Mississippi*	Newport News Co.	Cur. trb. grd. cr.	4	21	12.14	13.31	33.1	17.4
42	*Idaho*	New York S. Co.	Pars. trb. grd. cr.	4	21	28.75	30.88	45.1	35.3
	DESTROYERS:								
57	*Tucker*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	85.82	89.14	85.7	84.7
58	*Conyngham*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	87.54	91.03	88.4	87.6
59	*Porter*	Wm. Cramp & Sons	Pars. trb. grd. cr.	2	29.5	84.47	87.72	85.3	84.5
61	*Jacob Jones*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	93.04	94.63	92.2	92.2
62	*Wainwright*	New York S. Co.	Pars. trb. grd. cr.	2	29.5	92.23	93.83	91.5	91.5
63	*Sampson*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	70.64	71.10	65.1	60.5
64	*Rowan*	Fore River S. Co.	Curtis trb. grd. cr.	2	29.5	70.64	71.10	57.5	51.5
65	*Davis*	Bath Iron Works	Pars. trb. grd. cr.	2	30	45.92	55.68	57.3	54.0
66	*Allen*	Bath Iron Works	Pars. trb. grd. cr.	2	30	44.00	54 04	53.8	50.0
67	*Wilkes*	Wm. Cramps & Sons	Pars. trb. grd. cr.	2	29.5	48.52	51.42	34.1	28.6
68	*Shaw*	Navy Yard, Mare Isl'd	Pars. trb. grd. cr.	2	29.5	11.80	14.00	12.5	7.8
	FUEL SHIPS:								
14	*Maumee*	Navy Yard, Mare Isl'd	Diesel	2	14	73.82	75.31	96.8	96.5
15	*Cuyama*	Navy Yard, Mare Isl'd	Reciprocating	2	14	3.02	5.00	8.1	0.0
	SUBMARINE TENDER:								
2	*Bushnell*	Seattle Con. & D. D. Co.	Pars. trb. gearing	1	14	98.52	98.52	98.7	98 5
	DESTROYER TENDER:								
2	*Melville*	New York S. Co.	Pars. trb. gearing	1	15	98.26	99.37	99.8	99.8
	Transport	Navy Yard, Phila.	Reciprocating	2	14	12.76	23.83	36.1	33.0
	Supply ship	Navy Yard, Boston	Reciprocating	2	14	15.88	17.74	31.1	28.5
	TUGS:								
17	*Wando*	Navy Yard, Charleston	Recip. oil fuel	1	11	...	...	...	...
18	*Pocahontas*	Navy Yard, Norfolk	Recip. oil fuel	1	11	12.00	14.00	...	...
	Ferry Launch	Navy Yard, Charleston	Reciprocating	...	...	....	....	...	...
	Freight Lighter	Navy Yard, Portsmouth		...	...	90.00	90.00	...	...
	SUBMARINES:								
31	*G-3*	Navy Yard, N. Y.	Diesel-Sulzer	2	14	93.00	93.00	88.6	88.4
40	*L-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.9	98 9
41	*L-2*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	98.41	98.41	98.5	98.5
42	*L-3*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	96.89	96.89	98.1	98.1
43	*L-4*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.99	95.99	98.1	98.1
44	*L-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	14	19.90	30.70	81.0	74 5
45	*L-6*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	13.64	15.62	70.1	67.1
46	*L-7*	Lake, Long Beach, Cal.	Diesel-Sulzer	2	14	13.48	14.43	67.4	63.7
47	*M-1*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	95.43	95.56	88.7	88.4
48	*L-8*	Navy Yard, Portsmouth.	Diesel-Sulzer	2	14	4.57	5.05	56.2	53.7
49	*L-9*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	92.53	94.22	86.4	84.7
50	*L-10*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	91.96	92.11	83.9	80.4
51	*L-11*	Elec. Boat Co., Quincy	Diesel-New Lond	2	14	88.37	90.64	80.3	75.1
52	*Schley*	Elec. Boat Co., Quincy	Diesel-New Lond	2	20	...	....	0.0	0.0
53	*N-1*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	18 75	26.12	34.3	28 8
54	*N-2*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	18.75	24.59	34.3	28.8
55	*N-3*	Elec. Boat Co., Seattle	Diesel-New Lond	2	13	18 63	23.27	34.3	28 8
56	*N-4*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6.00	44.0	36.2
57	*N-5*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6 00	41.9	35.3
58	*N-6*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6.00	40.0	33.4
59	*N-7*	Lake Co., Bridgeport	Diesel-Sulzer	2	13	6.00	6 00	40.4	33.7

GERMAN SUBMARINE CONSTRUCTION IN BELGIUM.

The French journal "La Nature" reproduces a statement from the Dutch newspaper "Telegraaf," according to which the keels of nine submarines have been laid down at Hoboken, near Antwerp, since March 19. The men employed in the construction of the boats number 800. The main entrance to the yard is on the northern side. Since the aerial attack carried out by the British, the Germans have taken measures against a renewed attack. Above two parts of the yard they have built a roof formed of steel plates, covered by sand bags. The yard is, further, closed in by a thick wall provided with iron doors which can be instantly closed from the inside. By these means, the boats and the men are efficiently protected against any bombardment. There is, moreover, no risk of a fire spreading to the yard, and this is important, since two very large tanks containing naphtha and engine oil are close by. The smaller of these tanks is on the northern side, the other is to the south and in the ground; it measures 8 m. (26 feet) in length, 6 m. (19 feet 8 inches) in width, and 4 m. (13 feet) in depth. Between this and the river are located two buildings, the canteen, and the ambulance. There is, further, a small bay between the "naval yard" and the Cockerill yard, which the Germans have almost entirely covered over by means of tree trunks tied together, so as to afford a passage to the Cockerill yard. They have found use for a building called "La Chapelle," in which they have mounted anti-aircraft guns. The hulls of submarines alone are built at Hoboken; the whole of the machinery and outfit is manufactured in Germany.—"Engineering."

THE JAPANESE NAVY.

Japan contemplates a new naval construction expenditure of £2,333,332 in 1915–16. For 1916–17 the estimate is £3,604,849; for 1917–18, £2,681,864; and for 1918–19, £636,047. The total expenditure proposed in four years is considerably over £9,000,000. The naval estimates for the current financial year have been approved by the Japanese Diet. The estimates will defray in part the cost of three battleships of 30,000 tons each—one, to be named the *Yamashiso,* to be built at Yokohama; another, the *Ise,* to be built at the Kawasaki yards; and a third, the *Hyuga,* built at the Mitsu Bishi yard. The estimates will further partly provide for the cost of four first-class torpedo-boat destroyers of 1,000 tons each, two of which are being built in England, as well as for the cost of four second-class torpedo-boat destroyers and two submarines of 700 tons each. The keel of the *Ise* was laid on May 5, and that of the *Huyuga* on May 11. The three battleships are to be pushed forward vigorously, and the eight destroyers and the two submarines are to be completed by the close of 1916. The two torpedo-boat destroyers which are being built in England are to be named the *Urakaze* and the *Kawakazi.*—"Engineering."

SHIPBUILDING.

Lloyd's annual register of shipbuilding for 1914–15 is a further indication, if such be needed, of the amount of wealth which the war is sending into the United States in particular, and neutral countries in general. Obviously the construction of merchant vessels in this country has fallen off very considerably owing to the demands upon our shipbuilders for vessels of various kinds for Government purposes. In some yards the construction of ships for ordinary purposes has ceased altogether, and

the trade has been diverted elsewhere. Similar conditions, of course, apply to all countries engaged in the war, hence the great prosperity in shipbuilding circles of the United States, Japan, the Scandinavian countries and Holland. In the former there is under construction for classification under Lloyd's register, at the present moment, the largest amount of tonnage on record for that country. One of the not least interesting items in the report is the information that 22 new oil vessels, having a tonnage of 120,324, have been classed at Lloyd's, and many schemes are under consideration for still further increasing the quantity of tonnage for carrying oil by converting existing vessels, and those building, into tank steamers. The lack of tank steamer accommodation had always been put forward as the reason why prices of fuel and other oils were inclined to soar. With the greater demand upon the oil resources of the world at the moment, considerable impetus has been given to increase the tonnage of this class of vessel; indeed, dire necessity has forced it. In other directions there has been a heavy demand for vessels with refrigerating apparatus, and reconstructions and modifications have been made in vessels not so provided in order to meet this demand. The total number of vessels completed and classed with Lloyd's last year amounted to 733, representing 1,715,500 tons, which is slightly in excess of the figures for the previous year; but, as already pointed out, the distribution of this among the various shipbuilding yards of the world has undergone a startling change.—"Mechanical World," Manchester, England.

AN APPRECIATION OF A WELL-KNOWN ENGINEER.

CHIEF ENGINEER BENJAMIN F. ISHERWOOD.

The August number of our JOURNAL contained a tolerably full and appreciative obituary notice of this eminent man; but there were necessarily omitted several points of great interest to marine engineers in general, and those of the United States Navy in particular, and when his life shall be adequately written, all such notes will be of value: for this reason the following additional record is submitted.

It is not generally known that Mr. Isherwood, in his comparatively early days, wrote several articles for the "Franklin Institute Journal" of Philadelphia, when that periodical was almost the sole representative in this country of scientific and technical literature, and his translations from the French ranged through a long and busy life. Shortly before our Civil War he devoted great attention to the theoretical side of steam engineering, and kept in touch with the writings of Tyndall, Joule, Clausewitz, Heimholtz, Mayer, Thomson, Pouillet and other Europeans who showed that there was more to the subject than the mere boiling of water; even in the busy times of the war he caused some of the most notable of such writings in foreign magazines to be carefully copied, solidly bound and conspicuously labelled with the then somewhat mysterious title, "Thermodynamics." Perhaps his best known contributions to marine knowledge were included, besides others, in two volumes of "Engineering Precedents," referring to various vessels, experiments with propelling instruments, comparisons of coals, the expansion of steam, description of boilers, etc., and followed by two larger volumes of "Experimental Researches," in further pursuance of the same or similar subjects. When in after years the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS was suggested, Mr. Isherwood took great interest in the project,

wrote occasional articles and translations for its pages, and, as was fitting, he was given the place of honor in the first number of the first volume.

In the trying times produced by the Civil War Mr. Isherwood was conspicuous; for while it is true that the Navy contributed its full share toward the final victory of the Union arms, it may also be claimed that the Navy owed much of its efficiency to his efforts. As each problem presented itself he helped to solve it. The first requirement was for small armed vessels for coast and harbor service, and this call was partially met by the "ninety-day gunboats;" the second step was to furnish fast vessels for action in rivers and inlets, and which, to save maneuvering in narrow waters, could proceed in either direction; this need was fulfilled to a great extent by the "double-enders;" the third demand, toward the end of the war, was for excessively speedy ships to pursue privateers; this want was sought to be redeemed by very large ships driven by the "hundred-inch engines," and at such unprecedented naval speed for those days that some foreign journals refused to believe the authenticated accounts. When certain American engineers criticised his productions (and every man who achieves distinction will be criticised), he vanquished them by allowing them similar hulls to be propelled by machinery of their own design.

Perhaps Mr. Isherwood's chief characteristic was accuracy, he spared no pains for himself or his subordinates in the pursuit of truth by comparison and experiment, and the results, however voluminous, were almost always copied for publication by infinite labor in his own neat and distinct handwriting. He appears to have had time for few amusements, but he could unbend on occasion, and thoroughly enjoyed a good opera.

He was a man of mark, and when the time shall again come of peril to this country, those who are entrusted with the design, production and working of the machinery of our Navy, can do no better than imitate the energy, industry, forethought and patriotism of Benjamin F. Isherwood.

—F. G. McK.

BOOKS RECEIVED.

SHIP FORM, RESISTANCE AND SCREW PROPULSION. A book by G. S. BAKER, for the use of Naval Architects, Engineers and Draftsmen.

The subjects are handled from a practical viewpoint and theory is only introduced when it has a direct practical bearing. In Part I are nomenclature, stream-line motion, stern-friction resistance, eddy making, waves and wave making, ship-model experiments, curves of area, etc. In Part II, entitled "The Screw Propeller," the following subjects are taken up: screw propeller, elements of propulsion, hull efficiency, wake and thrust deductions, cavitation, etc.

Diagrams illustrative of the text are included.

Published by D. VAN NOSTRAND COMPANY, New York. Price, $4.50, net.

ENGINEERING PROBLEMS, by W. M. WALLACE, contains a collection of the most important rules and data on which are based the more usual calculations in engineering work, with problems to illustrate the application of these rules.

THE TECHNICAL PUBLISHING COMPANY, LTD., 55 and 56 Chancery Lane, London, W. C., England. Price, 3 shillings, net.

HYDRAULICS, by W. M. WALLACE. A treatise for engineering students and engineers in practice.

THE TECHNICAL PUBLISHING COMPANY, LTD., 55 and 56 Chancery Lane, London, W. C., England. Price, 4 shillings, net.

ASSOCIATION NOTES.

The annual meeting of the Society was held on October 5, 1915.

The following nominations were made for officers for 1916:

For President, Captain C. W. Dyson, U. N. Navy.

For Secretary-Treasurer, Lieutenant A. T. Church, U. S. Navy.

For Members of Council:

Captain T. W. Kinkaid, U. S. N.
Engineer-in-Chief C. A. McAllister, U. S. C. G.
Naval Constructor Robert Stocker, U. S. N.
Lieut. Commander Arthur Crenshaw, U. S. N.
Lieut. Commander J. H. Tomb, U. S. N.
Lieut. Commander H. T. Winston, U. S. N.
Lieut. Commander J. O. Richardson, U. S. N.
Lieut. W. T. Conn, U. S. N.
Lieut. S. C. Hooper, U. S. N.
Lieut. W. T. Lightle, U. S. N.

The election will be held on December 30, 1915.

It was decided to submit the question of giving a banquet during the year 1916 to a vote of the members.

The following members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Bean, Carlos, Lieutenant, U. S. N.
Boyd, Harry L., Captain of Engineers, U. S. C. G.
Dalton, Donald McL., Ensign, U. S. N.
Downey, Thomas F., Ensign, U. S. N.
Grove, George W., Ensign, U. S. N.
Hamlet, Harry G., Captain, U. S. C. G.

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Kinne, Merrill T., Ensign, U. S. N.
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VOL. XXVII. NO. 4.

JOURNAL

OF THE

AMERICAN SOCIETY

OF

NAVAL ENGINEERS.

PUBLISHED QUARTERLY BY THE SOCIETY, FOR THE ADVANCEMENT OF NAVAL ENGINEERING.

NOVEMBER, 1915.

WASHINGTON, D. C.

R. BERESFORD, PRINTER, 605 F STREET, N. W.

Entered at the Post Office at Washington, D. C., as Second Class Matter.

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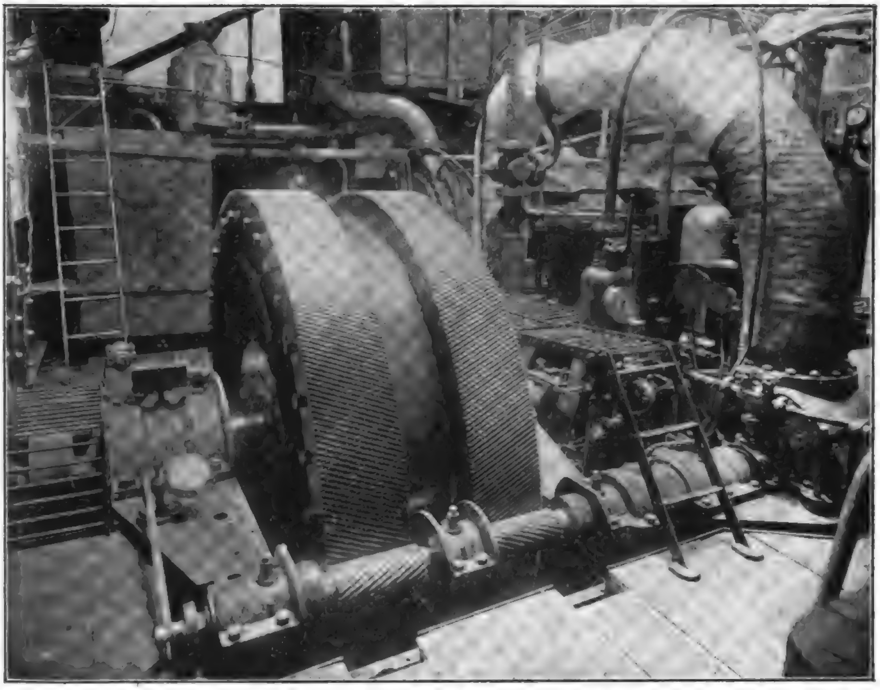

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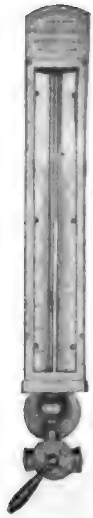

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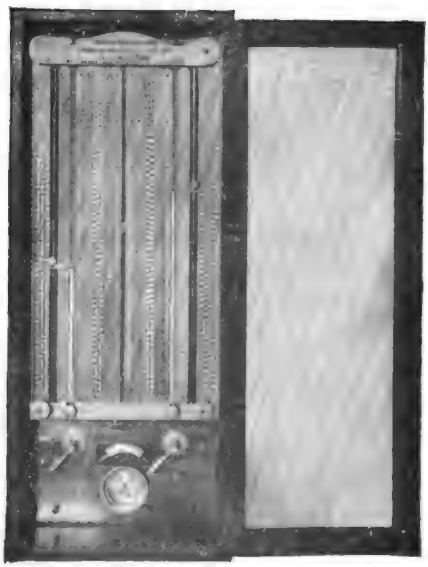

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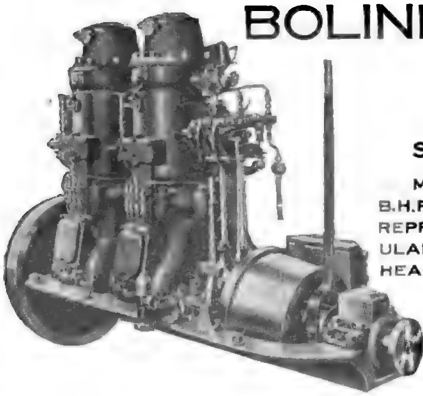

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